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THE ENGINEERS' DIGEST

REVIEW OF ENGINEERING PROGRESS ABROAD

PRINCIPAL CONTENTS

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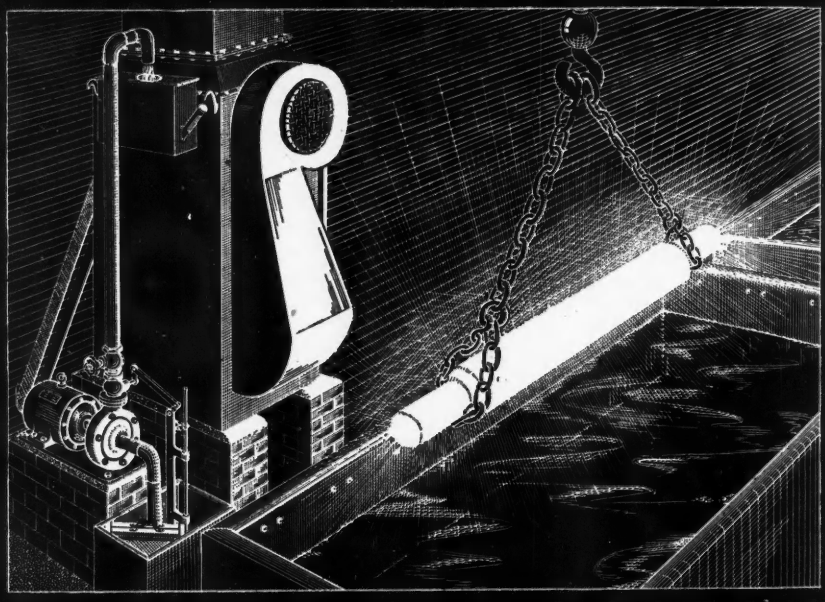

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EDITORIAL

The Air has become the arena in which the result of this war will be decided. Air power has become the key to ultimate victory. Supremacy in numbers must be secured together with supremacy in quality.

The latter requirement can be satisfied only by research work in the factories. The methods of this Industrial research are different from those adopted by the Physical Laboratories which work on a broad basis on fundamental questions. It has to be directed to a definite purpose, and is very often limited by time. Its results are to be found with an eye towards ease of application even if the theoretical problem encountered cannot be fully answered.

Commencing with this issue it is our intention to publish Reports of eminent German Aircraft Works so that we can give a fairly comprehensive picture of present German Industrial research. The publication of such Reports should allow a most useful comparison with similar activities of our Industrial Establishments.

* * *

All future issues of THE ENGINEERS' DIGEST *will be printed on Art paper. This will greatly improve on the reproduction of Diagrams and Photos, and we are very grateful for the increasing reader-support which has made this improvement possible.*

* * *

Abstracts from the principal Russian technical journals, many of which have very high standard, will also be included in future issues of THE DIGEST. *They will be not only of topical interest but will contribute to a better knowledge of the amazing development of Russian Engineering in recent years. The aim in the selection of the articles will be to give our Readers information of practical use.*

VACUUM PRODUCTION IN CONTAINERS BY ROTARY PUMPS

By Dipl. ING. TH. BREMI, SLM., Winterthur. (From *Schweizerische Bauzeitung*, Vol. 116, No. 4, July 27th, 1940, pp. 37-41).

ROTARY pumps are very suitable for producing vacuum, since due to their continuous action, if running at comparatively high speed, no pressure waves or shocks are transmitted to the container or pipe line. Three types of rotary pumps most suitable for this purpose are illustrated in Fig. 1. According to the conditions and requirements such as vapour pressure, etc., one or other type of pump is employed.

a b c

Fig. 1

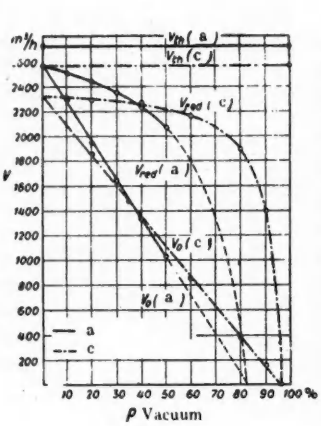


Fig. 2

The principle task in designing the plant is to determine the time required for producing the required vacuum. In the following this will be calculated analytically. The same principle applies to all three types of pump, as can be seen from Fig. 2.

Pump Characteristics and General Equation for the Time of Evacuation.

According to Fig. 3, the volume to be evacuated V_{red} (theoretical), reduced to the condition at the suction valve (pressure and temperature), from $0 + 100\%$ is constant. Hence the suction volume reduced to atm. pressure and temperature can be represented by a straight line indicated by V_{red} (Fig. 3) from $p=1$ atm., to V_o at $p=0$. This applies also to the actual machine where, due to leakage, V_{red} at $p = 1$ atm. is replaced by V_{max}. The curve of the suction volume $V_{red} = V \frac{p_o}{p}$ is of hyperbolic form.

From Fig. 3

$$V_o = V_{max} \frac{p - p_2}{p_o - p_2}$$

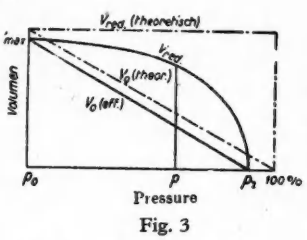


Fig. 3

where p_2 is the limiting vacuum which can be obtained with the pump, and V_{max} represents the suction volume of the pump at pressure p_o.

Hence

$$V_{red} = V_{max} \frac{p - p_2}{p_o - p_2} \cdot \frac{p_o}{p}$$

The weight of gas pumped out per revolution is

$$W = \frac{PV}{RT} \text{ i.e.}$$

$$W = V_{max} \frac{p - p_2}{p_o - p_2} \frac{p_o}{RT}$$

In most cases the temperature remains constant in the container, so that the product RT can be regarded as constant. Hence the quantity of gas extracted during a small interval of time dt, is given by

$$dW_1 = V_{max} \frac{p - p_2}{p_o - p_2} \frac{p_o}{RT} dt.$$

If air is entering the container during the evacuation (due to leakage) through an opening of cross-sectional area A, then the gain in weight in the container during the same time interval dt, is given by

$$dW_2 = A\,\psi\,dt, \text{ where } \psi = f(p).$$

The coefficient of contraction of the area (μ) is taken into account if $A = \frac{A'}{\mu}$ where A' represents the actual cross-sectional area of the opening.

The weight of gas in the container is given at any instant, by

$$W = \gamma J_p = \frac{J_p \cdot p}{RT}$$

where J_p is the volume, which can vary with the decreasing pressure as will be seen later.

The reduction of weight during the time interval dt, corresponding to a small change in the pressure dp is given by

$$dW_3 = \frac{1}{RT}\,[\,d\,(J_p \cdot p)\,] = \frac{1}{RT}\,\frac{\delta\,(J_p \cdot p)}{\delta\,p}\,dp.$$

Equating dW_3 with $dW_1 - dW_2$ and solving the equation for t, we get

$$t = \int_{p_o}^{p} \frac{(J_p \cdot p)'}{V_{max}\,\frac{p - p_2}{p_o - p_2}\,p_o - \text{F.R.T.}\,\psi}\,dp + c.$$

Case 1, Closed Vessel.

In this case $A = O$, and J_p = constant, further c can be evaluated by putting $p = p_o$, for $t = 0$.

$$\text{Hence, } t = \frac{J_o}{V_{max}}\,\frac{p_o - p_2}{p_o}\,k.\ 1.\ \frac{p_o - p_2}{p - p_2}$$

See also fig. 4.

Case 2, Container with Air leakage through an Opening of Cross-Sectional Area A.

In this case the obtainable vacuum is not only limited by the quality of the pump, but also by the value of p_2, which in this case is increased to p_{min} due to leakage. At the beginning of pumping air is leaking into the container under the critical pressure p_c, where

$$p_c = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \cdot p_o = 0.5282 \cdot p_o$$

after the critical pressure is reached the air enters the container above critical pressure, i.e., the inward flow is constant. Hence the time for evacuation can be calculated in two parts:

(A) With under critical pressure inflow

$$dW_2 = \sqrt{2g\,\frac{n}{n-1}\left[\left(\frac{p}{p_o}\right)^{2/n} - \left(\frac{p}{p_o}\right)^{\frac{n+1}{n}}\right]}\,A \cdot dt$$

from which

$$t_1 = \frac{J_o}{V_{max}} \int_{p_o}^{p_c} \left\{ p - \left[p_o \sqrt{RT}\,\sqrt{2g\,\frac{n}{n-1}} \cdot \frac{A}{V_{max}} \cdot \sqrt{\left(\frac{p}{p_o}\right)^{2/n} - \left(\frac{p}{p_o}\right)^{\frac{n+1}{n}}} \right]^{-1} \right\} \cdot dp + c$$

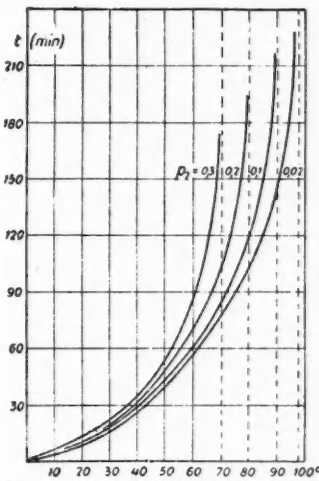


Fig. 4

In this equation $p_2 = 0$ is substituted, since p_2 is negligible compared with p or p_o.

The analytical solution of this equation is not given, but the graphical solution (left half of Fig. 5) gives good results. For this purpose

$$t_1 = \frac{J_o}{V_{max}}\,t_o$$

is calculated for $t_o = f(p)$, with various values for the parameter $\frac{V_{max}}{A}$

(B) With air inflow above critical pressure:

$$dW_2 = \left(\frac{2}{n+1}\right)^{\frac{1}{n-1}} \sqrt{\frac{2gn}{n+1}} \cdot \frac{p_o}{\sqrt{RT}} A \cdot dt =$$

$$= 2.15 \frac{p_o}{\sqrt{RT}} A \cdot dt$$

and hence,

$$t_2 = \frac{J_o}{2.15 \sqrt{RT}} l.k \left| \frac{\frac{V_{max}}{A} \frac{(p-p_2)}{2.15 \sqrt{RT}\,(p_o-p_2)} - 1}{\frac{V_{max}}{A} \frac{(p_c-p_2)}{2.15 \sqrt{RT}\,(p_o-p_2)} - 1} \right|$$

c being evaluated by putting $t = 0$ for $p = p_c$ in the equation.

The right hand side of Fig. 5 shows the decrease in the limiting vacuum due to air leakage. The total time required for evacuation is the sum of t_1 and t_2.

Case 3, Liquid Entering the Container.

This case is investigated with water entering the container from a constant height, as in case of priming a water pump (Fig. 6). Naturally the volume of the container above the water level is changing continuously, hence J_p takes the place of J_o, J_p being the value corresponding to pressure p in the container. Also the time required to evacuate the container is influenced by its shape, hence a general solution is not possible. Three cases are investigated: vertical cylindrical vessel, horizontal cylindrical vessel, and a spherical vessel. Results for vessels of different shape can be obtained by interpolation of the results of the above mentioned three cases.

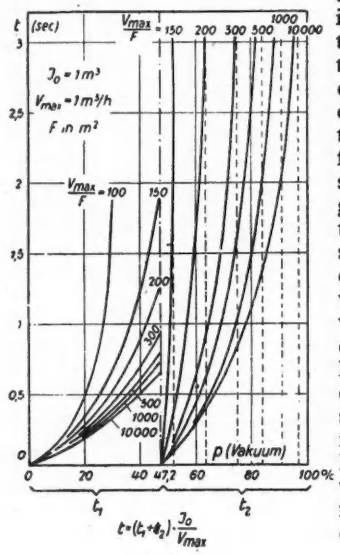


Fig. 5

(1) **Vertical Cylindrical Container.**

From Fig. 6 the value of h as a function of p, the pressure above the water level, can be determined as:

$$h = h_o \frac{p_o - p}{p_o - p_1} \text{ and hence } J_p = J_o \frac{p - p_1}{p_o - p_1}$$

hence,

$$t = \frac{J_o}{V_{max}} \frac{1}{p_o} \frac{(p_o-p_2)}{(p_o-p_1)} \cdot$$

$$\cdot \left[(2p_2 - p_1)\, l \cdot k \frac{(p_o - p_2)}{(p - p_2)} + 2\,(p_o - p) \right]$$

c again is evaluated by putting $t = 0$, for $p = p_o$.

(2) **Spherical Container.**

Water enters the container at the lowest point, and suction takes place at the highest point. Then

$$J_p = \frac{\pi}{6} (h_o^3 - 3 . h_o . h^2 + 2 . h^3) \text{ or substituting}$$

$$h = h_o \frac{p_o - p}{p_o - p_1}$$

$$J_p = \frac{J_o}{(p_o - p_1)^3} (3 . p_o . p_1^2 - p_1^3 - 6 . p_o . p_1 . p + 3 . p_1 . p^2 + 3 . p_o . p^2 - 2 . p^3)$$

An example is solved graphically. In the equation

$$t = - \frac{p_o - p_2}{p_o} \frac{1}{V_{max}} \int_{p_o}^{p} \frac{(J_p . p)'\, dp}{(p - p_2)}$$

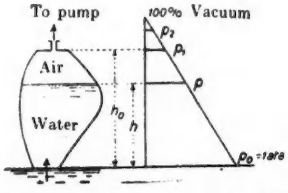


Fig. 6

the value of $J_p . p$ is first graphically differentiated and following that, the whole expression is graphically integrated. Fig. 7 shows the solu-

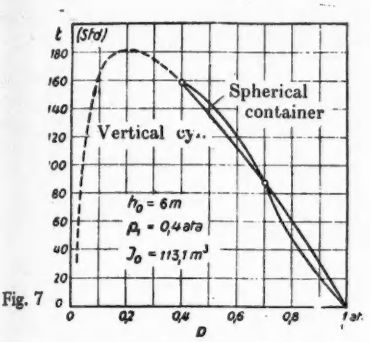


Fig. 7

tion for an example where $H_o = 6$ m. and $V_{max} = 1$ m³.

(3) **Horizontal Cylindrical Container.**

The length of the cylinder is L, and the diam. of the cylinder is H_o, hence

$$J_p = \frac{h_o^2 L}{8}\left[2.\pi - 2\cos^{-1}\left(1-\frac{2h}{h_o}\right) - \frac{4}{h_o}\sqrt{h\,(h-h_o)}\left(1-\frac{2h}{h_o}\right)\right]$$

Substituting $h = h_o \dfrac{p_o - p}{p_o - p_1}$ hence,

$$J_p = J_o - \frac{h_o^2 L}{8}\left[2\cos^{-1}\left(\frac{2p - p_o - p_1}{p_o - p_1}\right) - \frac{4}{h_o (p_o - p_1}\sqrt{h_o\,(p_o - p_1)\,(2p - p_o - p_1)}\right]$$

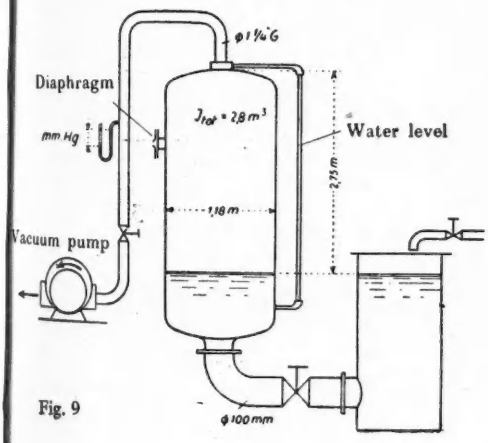


Fig. 9

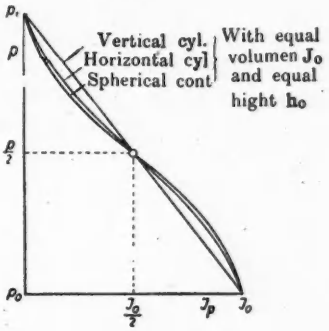


Fig. 8

A quick method to solve these equations is shown in Fig. 8, where values for J_p are plotted as a function of p, for the three types of containers.

Experiments to Check Theoretical Results.

Checking experiments were carried out with the arrangement shown in Fig. 9. The results being as follows:

Evacuation of a closed vessel with a pump, for which $V_{max} = 52.1$ m³/h and $p_2 = 0.05$ atm. at $p_o = 0.988$ atm. was predetermined.

The time of evacuation was calculated and checked by experiment:

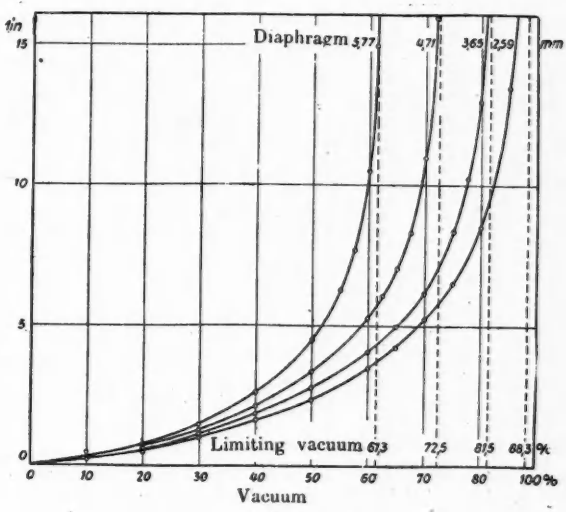


Fig. 10

Vacuum	%	40	60	70	80	85	90
Calculated $t=t_{th}$	min.	1.73	3.19	4.27	5.93	7.23	9.45
Measured t	min.	1.55	3.15	4.22	6.06	7.70	10.67

Evacuation of a vessel with air leakage.

Diam. of opening	mm.	2.59	3.65	4.71	5.77
Value of $\frac{V_{max}}{A}$	m/h	1000	500	300	200
Limiting Vacuum	%	88.3	81.5	72.5	61.3

The results are shown in Fig. 10.

Evacuation of a vessel with water inflow from a constant height, the pump could be adjusted to two values of limiting vacuum, by arranging the leakage losses of the pump—

Limiting vacuum of the pump p_z	atm.	0.08	0.177
Barometric pressure p_o	atm.	0.975	0.977
Suction volume V_{max}	m^3/h.	11.8	10.0

Time to fill the vessel with water

$p=p_1=0.707$ atm.	Measurement	min.	16.55	19.75
	Calculation	min.	16.65	20.10

From the results it can be seen that calculation and experimental results agree very closely.

AXIAL THRUST OF HIGH PRESSURE TURBO-PUMPS

By K. RUTSCHI. (From *Schweizerische Bauzeitung*, Vol. 117, No. 16, April 19th, 1941, pp. 176-177).

THE maximum head against which a single stage turbo-pump can deliver is limited by the speed of the runner. Therefore, if the head is very large, several pump stages are connected in series. If the fluid is admitted to each stage from the same side, as is usually the case, an end thrust exists which has to be taken up by thrust bearings and glands or be balanced otherwise.

In Fig. 1 the axial thrust is taken up by a thrust block and is transmitted to the pressure on the inlet side. With this design the gland behind the thrust block is free from pressure, but owing to the fact that some water is carried back to the suction ports, there will be a loss of efficiency. When the pump is dealing with hot water steam might leak through the thrust block. Another possibility is the thrust block seizing on the shaft causing an axial displacement of the shaft and consequently of the impeller. This defect might be overcome by using thrust bearings in place of the blocks.

The axial thrust may be balanced by a method illustrated in Fig. 2, where the working liquid enters the runner at both sides, so that the end thrusts on the impeller oppose each other. With this design no pressure is exerted on the glands. Owing to the parallel operation of each two corresponding stages a large number of pump units have to be used if the head is large. Fig. 3 illustrates another solution; here, however, all the stages are in series. The gland on the pressure side of the pump is exposed to half the end thrust. The design shown in Fig. 4 is a distinct improvement regarding axial thrust equalisation, pressure on glands and maximum pressure drop in the stages. The unfavourable disposition of the complex piping makes the design complicated and limits the number of stages that can be arranged for. This necessitates individual design for every pump.

Recent designs originated by German engineers endeavour to solve the problem on entirely new lines. The schematic arrangement is shown in Fig. 5; it is essentially a multi-

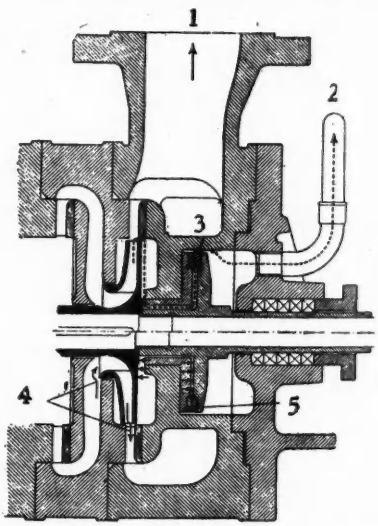


Fig 1

(1) Delivery Volumen Q; (2) Loss of liquid by relieving pressure; (3) New collar rings; (4) Flow of liquid broken; (5) Collar rings worn out.

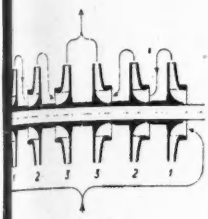


Fig. 2

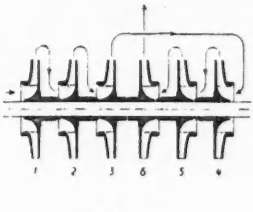


Fig 3

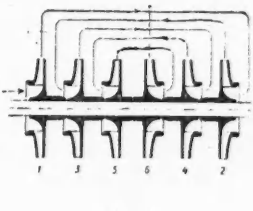


Fig. 4

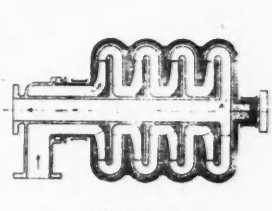


Fig. 5

stage pump connected rigidly with a steam turbine so that the axial thrusts of the turbine and the pump are opposed. To eliminate a pressure gland, the casing rotates with the runners on a hollow shaft which serves as the suction branch. In another design the pump runners are shifted inside the turbo runners and is especially favourable regarding weight and space economy. The high temperatures and speed necessitate a very careful design and considerably increases the manufacturing costs. No data are available as to whether the plant came up to expectation. At any rate the gain by eliminating the use of thrust bearings or blocks is balanced by increased losses through friction, since the dimensions of the parts rotating in water are increased by about 25 to 40% and this increases considerably the frictional losses.

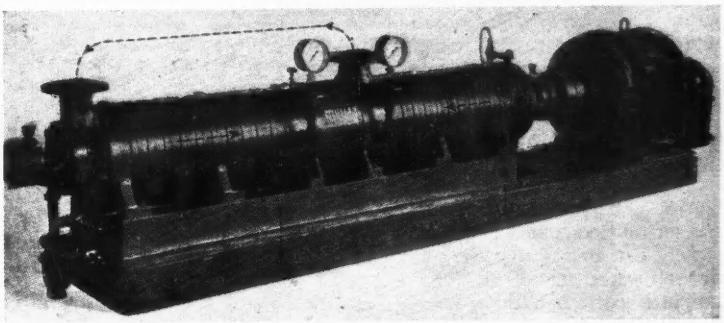


Fig. 8

A new design has been developed by Messrs. Müller A.G., and is shown in Fig. 6. Two

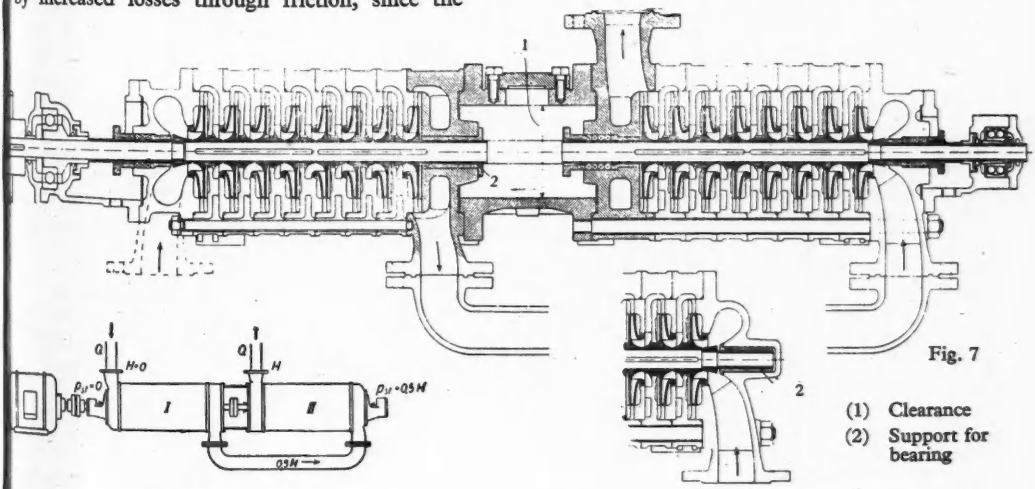


Fig. 7

(1) Clearance
(2) Support for bearing

Fig. 6

Fig. 7a

pumps are rigidly connected, the connection being covered by a casing closed to the atmosphere, hence there is no need for open glands at the coupling side. The two pumps are so arranged that their end thrusts are opposed. The direction of flow in the pump is indicated in Fig. 7; and corresponds to that shown in Fig. 3. The gland at the pressure side of the second pump is under half the end thrust, but this can be completely eliminated by internal bedding of the shaft as shown in Fig. 7a.

Fig. 8 shows a pump of this type with an output of 5 l/sec. against a head of 540 m. For high efficiency 24 runners of small diameter are used. In spite of the large number of stages it is well designed and runs very smoothly at 2900 r.p.m. The pumping characteristics of this unit are shown in Fig. 9. Dotted lines indicate the guaranteed figures, and full lines the actual values obtained during test running. The efficiency, which is 69% (2% in excess of the guaranteed efficiency), is very high, especially considering that the pump was designed for hot water service and the clearances were designed correspondingly larger, causing increased losses when pumping cold water as during the test.

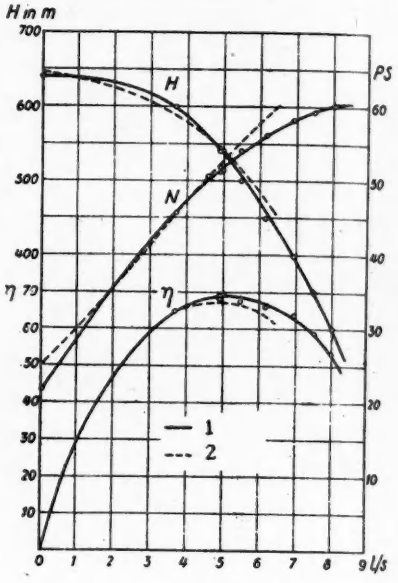


Fig. 9
(1) Test; (2) Guarantee

NEW SWISS GRINDING AND MILLING MACHINES

By ALFRED RICKENMANN. (From *Schweizerische Bauzeitung*, Vol. 117, No. 16, April 19th, 1941, pp. 180-182).

AT the 1941 Basle Fair some new thread grinding and milling machines were introduced by the Swiss firm, Reishauer-Werkzeuge A.G., which are subsequently described.

A universal thread grinding machine, type NRK, is shown in Fig. 1. Owing to its great versatility it can be used for grinding threads of all descriptions, such as right and left hand, internal and external threads with a pitch of about 0.4 to 0.8 mm. The grinding wheel is adjustable from both sides up to about 25° helix angle. The machine can work with a single or a multi profile grinding wheel of 40 mm. maximum width. With a single profile grinding wheel it makes use of the longitudinal traverse grinding method, while with a multi profile wheel it operates by the plunge-cut method. The first method is used for spindles, worm gears, long screw taps, and thread gauges, that is whenever the length of the thread exceeds 40 mm., or when it would not pay to prepare a multi profile wheel. For short taps, or for grinding threads on bolts, screws, etc., made of high quality steel the second method is advantageous. It eliminates rough machining of the thread, and hence the adjustment of the grinding wheel in the rough thread.

The single profile wheel is formed by means of a diamond, while for the multi profile wheel crushing rolls are used which are pressed against the wheel. For precision work the machine is fitted with a hardened and ground grinding spindle and a special pitch correcting, change speed, gear. For easy setting of the grinding wheel in rough machined threads, a telescope with a revolving graticule is provided. The electrical machinery consists of four motors, all controlled by press buttons. The reversing of the slide is hydraulically operated, and so is the relief grinding motion of the slide.

Another new model introduced by the same firm is the R. 1 shown in Fig. 2. This machine

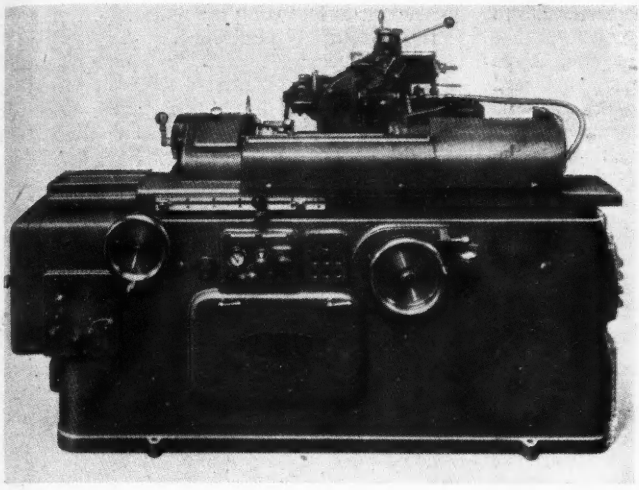


Fig. 1

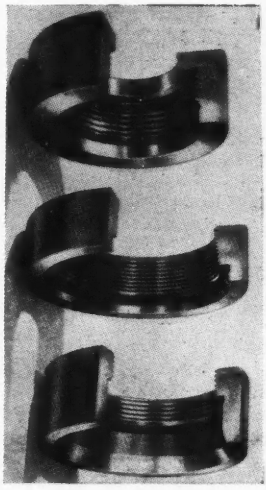


Fig. 3

is a combined Internal, Thread, and Hole Grinder with provision for dealing with parallel or taper holes. Examples of suitable work are shown in Fig. 3. Thread grinding is always performed with multi profile grinding wheels.

Since the speed of travel for thread and plain hole grinding is different, provision is made for a quick change over from one type of work to another. For this purpose two table slides are provided, one above the other, the lower one for the fast traverse when hole grinding being hydraulically operated, and the other one for thread grinding operated through gearing.

The speed of the machine can be steplessly varied between wide limits. Three internal spindles are provided with a range of speeds from 4,000 to 28,000 r.p.m. to amply cover the various grinding wheel diameters that will be required for the range of work of which the machine is capable. A tachometer indicates the spindle

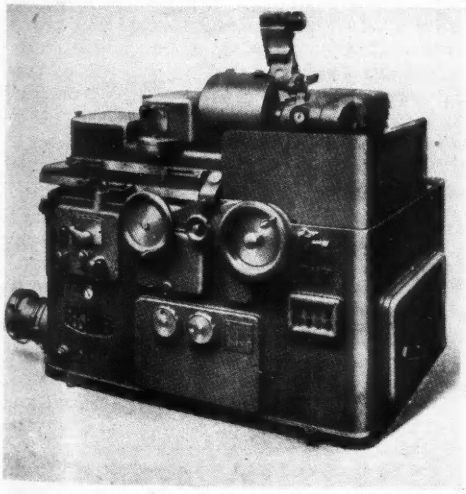


Fig. 2

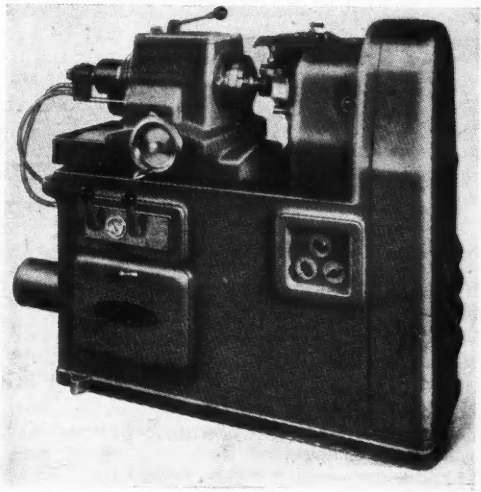


Fig. 4

speed. Work speeds are variable from 0.5 to 256 r.p.m. in 32 steps.

The short thread milling machine, Type KBH, shown in Fig. 4, is principally a thread groove milling machine. The work spindle provides the longitudinal and transverse motion for the pitch and the depth of thread. After the work is finished the spindle is brought automatically to its original position. Operation of the machine consists only of feeding the work to the machine and starting the motor. All other work is automatically performed.

The machine is especially suitable for cutting short internal and external threads. The following operations are hydraulically controlled; clamping and releasing of the work, advancing and returning the work slide, fixing the slide into position, and automatic operation of a milling cutter protection cover during changing of work.

WIRES AND CABLES WITH INSULATION BY VERY THIN FOILS

By WALTER FISCHER. (From ETZ *Elektrotechnische Zeitschrift*, Vol. 61, No. 8, February 22nd, 1940, pp. 163-65).

OF all types of insulated wires enamel coated wires have the thinnest insulation; it is no thicker than a few hundredths of a mm. This thin film is applied to the wire by drawing the cleaned conductor several times through the enamel material, with which it is to be coated, and fixing each coat by passing the wire through a drying chamber. The cover produced in this way is astonishingly ductile, and shows great resistance against scraping. According to the rules of the VDE (DIN VDE 6450) the enamel must not burst, when the wire is elongated by 25%; good products comply easily with this regulation. Besides, such wires can be bent round a mandrel with a diameter equal to that of the wire without a burst of the insulating film; this corresponds to an elongation of 50% of the surface fibre.

It has been found that this high ductibility and strength of the insulating cover is not a quality of the insulating material; but by the method of the manufacture the thin film is in such a perfect contact with the surface of the wire, that a cracking of the film cannot occur before the tensile forces in the insulating film at one point become greater than the molecular forces, which attach the film to the surface of the conductor.

This being so the same good mechanical qualities were anticipated in an insulating film produced on a wire by using a different process, which gives the insulation the same good adhesive property as the coating of the wire with the varnish producing the enamel. Great importance has been attributed to this problem, in order to be able to use some cellulose derivates as insulating materials, e.g., hydrocellulose, cellulosetriacetate, polystryol, polyvinyl-chloride, and others, which could not be applied to the wire by the method used for the enamel wire.

After extensive experiments a new practicable process was found.* Foil bands, 10 μ to 30 μ thick, one side coated with a 5 μ layer of an adhesive substance are wrapped round the conductor and affixed to the surface of the wire by the adhesive substance. The wrapping is done in such a way, that the insulating foil, spread on one side with the agglutinant, is introduced into a guiding mechanism, which wraps the band round the wire in a manner similar to that used in manufacturing insulated wires with a metallic sheath over the insulation (type of wire used in Germany, called "Rohrdraht"). The new process has been developed for wires up to to 0.8 mm. at first.

The speed of manufacture could be brought up to more than 2000 m/hour. It is possible to wrap the wire more than once, if the band has sufficient width. For practical reasons the number of wrappings is limited to about six at present, but there exist no fundamental difficulties for an increase of this figure, so that, e.g., with a foil band of 20 μ an insulation of 10 to 20 layers could be built up. The speed of manufacture does not depend on the number of wrappings, and is not influenced by the diameter of the wire.

Fig. 1 shows the enlarged cross section of a wire of 1.4 mm. diameter, insulated with two foils, each wrapped 4 times around the wire. There is no difference between these new wires and the enamel wires in regard of mechanical properties. But in one regard the new wires are even superior to the enamel wires. The rules of the VDE for enamel wires have limited the number of pinholes to 15 in 15 m. Good

* The experiments were carried out by the Deutsche Kabelwerke A.-G.

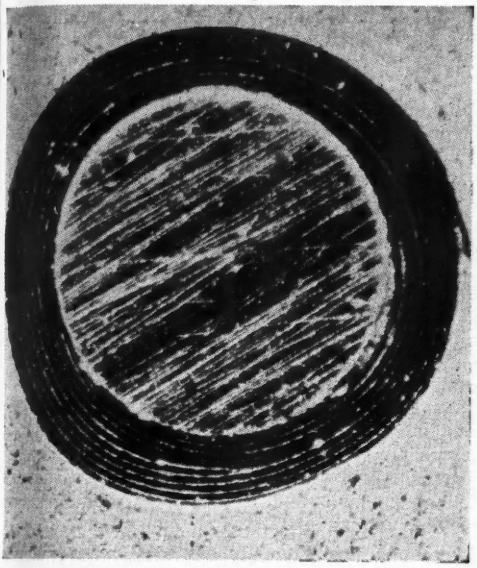


Fig. 1

Wire with foil-insulation, two films, each wrapped four times round the wire. Diameter of wire: 1.4 mm.

enamel wires have regularly less defects. With the new process, however, it is possible to manufacture wires without any pinholes, especially if the film is wrapped several times round the wire.

By the choice of films made from special materials insulated wires can be produced, similar to the enamel wires, but with qualities not possessed by the latter, as is shown by the following. Tests made with wires insulated with films of cellulose-triacetate (thickness of insulation 30 to 120 μ) gave the following results:

Dielectric strength to a metallic mandrel—higher than	700 V/10μ
Dielectric strength in water of room temperature in the first hour of immersion..	1000 V/10μ
Dielectric strength after 30*h.* of immersion about	200 V/10μ

This value is constant even with longer stay in water.

The resistance to heat is demonstrated by the following test: a wire is hung over a metallic mandrel of a diameter six times that of the wire. The ends of the wire are stretched by weights of 2 kg. Even at a temperature of 155° C. no impressions on the insulation will be found.

Such wires are most suitable for dynamo and other windings. If the windings are impregnated, one has to take care, that the materials of the film and of the impregnation agree together from the chemical point of view.

A film of cellulose-triacetate is very insensitive to water. If an extremely high degree of insensitivity is required, foils of polystryol or polyvinyl-chloride have to be used. These films have a lower softening point. But it is probable that a suitable combination of different foils will be found, which will fulfil any conditions which have to be taken into consideration.

The new films can be coloured easily. Insulated wires with bright colours can be produced, which are most suitable for switch board connections. The thickness of the insulation of such wires may not exceed 15 μ.

The following example will give an idea of what can be attained by the combination of two foils of different material. The problem was to give to a wire of 0.4 mm. an insulation of very good dielectric qualities and great resistance to humidity. The wires showed at the beginning of the test 4500 MΩ/km. and after being seven days in water of room temperature an insulating resistance of 3800 MΩ/km. The capacity between the wire and the water at the end of the seven days was 0.35 μ F/km. and tan δ was 0.05 with 800 Hz. The dielectric strength was 8 kV, according to a time-breakdown curve. When these measurements were finished the temperature of the water was raised to 60° C. After one hour the insulation resistance was found to be 500 MΩ/km.

THE DEVELOPMENT OF TWO-STROKE ENGINES FOR MOTOR CARS

By H. J. VENEDIGER. (From *Zeitschrift des VDI*, Vol. 85, No. 7, February, 1941, pp. 187-199).

THE modern two stroke engine is based principally on the two stroke, crank case compression, in line type engine, which has been improved considerably during the last few years. At present, a very favoured engine is the two cylinder, water cooled type, with a stroke volume of 300 to 500 cc. for each cylinder, and its design is so compact that it can be fitted into the car in an axial or transverse direction equally well. Multi-cylinder engines of this type are also made

without any difficulty. Among these, the three cylinder type is preferred to the four cylinder engine, because of its better balance and superior torque characteristics. Further development is expected to replace the water cooling by air cooling system.

Five cylinder engines of about 50 h.p. output with a firing order 1, 5, 2, 3, 4 or 1, 4, 3, 2, 5 have a very symmetrical crank shaft design and smooth running. Engines with more than 5 cylinders are of considerable importance as, in common with the 5 cylinder type, they are well balanced, especially when rotary blower is adopted.

The two stroke opposed piston type has so far only been used for experimental purposes. Since, in this case, two pistons are operating in conjunction with one crank case chamber, its torque diagram corresponds to that of a two stroke engine, with the same stroke volume but only half the number of cylinders. The advantages of the two stroke system are thereby nullified.

Engines with separate scavenge pumps for each cylinder, or one double acting pump for every two cylinders, could be designed with a much greater variety of cylinder arrangement if it were not for the complicated shape of the crank shaft. The adoption of double acting pump is necessarily limited to twin cylinder engines. Four or more cylinders, with double acting pumps, may be built as V or Star engines.

Consequently the following types are available for automobiles :

(a) Small cars. Two or three cylinder in line type, with crank case compression ; stroke volume 0.6 to 1.0 litres. Four cylinder, opposed piston, type with crank case compression stroke volume 1 to 1.2 litres. Three cylinder radial engine with separate scavenging pumps ; stroke volume 0.7 to 1.0 litre.

(b) Medium cars. Four or five cylinder, in line type engines, with crank case compression, 1.25 to 2.1 litres stroke volume. Four cylinder, opposed piston type, with crank case compression, 1.2 to 1.5 l. stroke volume. Four cylinder, V engines of 1.2 to 1.5 l. stroke volume, with single or double acting pumps. Five cylinder radial engines up to 1.75 l. stroke volume with five single acting pumps or a rotary blower.

(c) Large cars. Five, six and seven cylinder, in line type engines, of 1.75 to 2.5 l. stroke volume with rotary blower. Eight cylinder, double star engines, up to 3.1 l. stroke volume, with rotary blower.

Of these designs only a few have been produced on a commercial scale, namely, the two and four cylinder, in line type, with crank case compression, and the four cylinder, V type, with double acting pumps. The best known are the two cylinder, in line type, with 0.8 l. to 0.7 l. stroke volume (Fig. 1). The stroke and bore of these engines is 68/74 and 76/76 mm. respectively. The weight of the engine is 3 and 2.9 kg/h.p. respectively. As shown in Fig. 2 these engines may be built in transversally into the car.

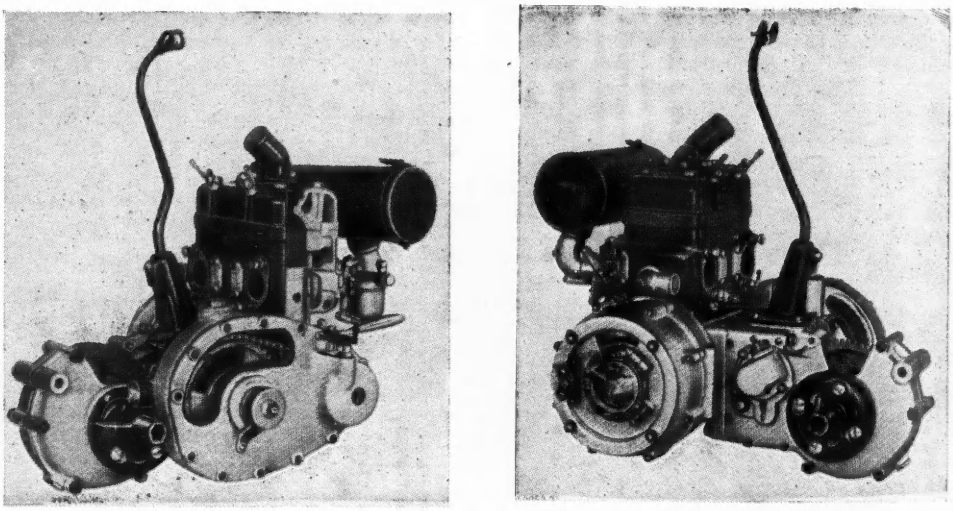


Fig. 1

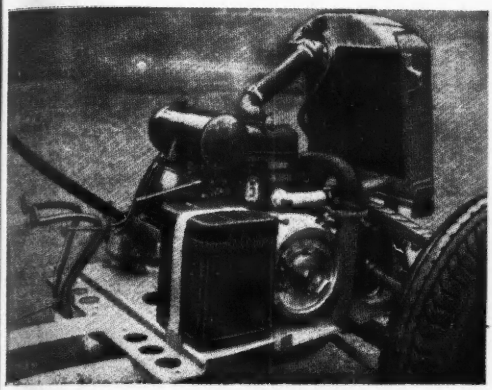


Fig. 2

In Fig. 3 a two cylinder type, manufactured in Czechoslovakia, has a stroke volume of 0.73 l., and an output of about 10 h.p.

Another type, the "Aero 30," also manufactured in Czechoslovakia, is shown in Fig. 4. This is a two cylinder engine of extraordinarily large dimensions, the stroke volume is 998 cc., the bore 85 mm., the stroke is 88 mm., and the output at 3200 revs. is 30 h.p. It is, however, doubtful whether this engine is capable of smooth running and continuous operation at full load.

Fig. 5 shows a 4 cylinder, V engine of 1060 cc. stroke volume. Its weight is 4.1 kg./h.p., and is characterised by a very smooth running over the entire speed range.

Up to the present, two stroke engines are built up from three parts: crank case, cylinder block, and cylinder head; for four stroke engines these parts are cast in one. Both these solutions may be justified for either type by strength and

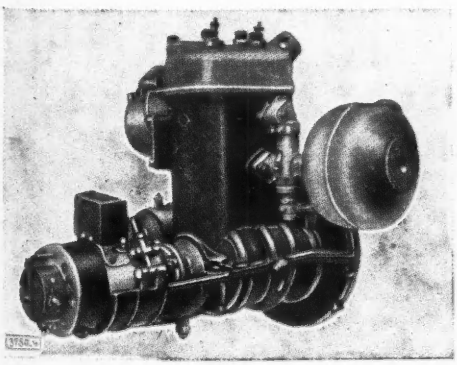


Fig. 3

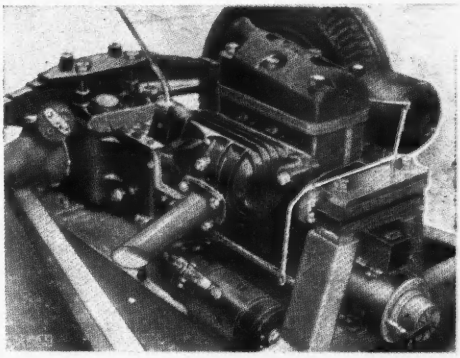


Fig. 4

simplicity of dismantling. For the two stroke engine the crank case is usually made of light metal, the cylinder block and cylinder head of cast iron. Air cooled engines are fitted with aluminium cylinder heads. The inlet and outlet parts in the cylinder block require accurate casting; if, in addition, finishing is needed, openings are left in the casting which are closed after finishing.

The dimensions of the ports must not exceed a certain value to prevent clamping of the piston rings in these ports. As a rule the maximum allowable circumferential length of the ports is about 6/10ths of the cylinder circumference, and any one opening must not exceed 1/5th of the cylinder circumference. It is advisable to use several narrow ports in place of a single larger one, to assure uniform expansion of the cylinder at high temperatures.

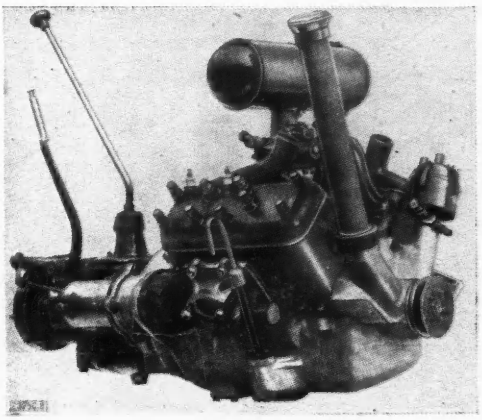


Fig. 5

Recently, to improve packing, aluminium cylinder blocks were used, on which a layer of steel was sprayed. This also improved the heat transfer and smoothness of running. The disadvantage of this method is that no satisfactory adhesion could be obtained between the aluminium and the steel.

The general tendency for two stroke engines, too, is to use a larger number of cylinders with a smaller stroke volume. At present, a bore of 78 mm. and a stroke volume of 400 cc. per cylinder may be regarded as the upper limits. Future engines will probably use cylinders of 65-75 mm. bore, with a stroke/bore ratio of 1.25.

Driving Mechanism.

Engines using mixed lubrication (fuel-oil mixture) are invariably fitted with roller bearings. The crank shaft and the connecting rod big end are directly or indirectly supported by the bearing according to the manufacturing cost, available space, and lubrication arrangement. With the direct bearing support, the rollers run in case hardened grooves (about 1.0 to 1.5 mm. deep), cut in the crank shaft; for indirect bearing support, inner and outer ball races are used, fitted in the connecting rod big end, or on the shaft. The wrist pin and the ball races are made of hardened chrome steel with 0.1 to 0.2% C., and 0.6% Cr., while the rollers are made of chrome steel, 1% Cr., and 1% C.

Preference is given to the direct bearing support, in the case of small cars, owing to the cheaper cost. Main bearings, other than the end bearings, are specially designed; an outer ball race is fitted into the crank case in which the rollers of the bearings are running directly on the crank shaft (semi-direct bearing support).

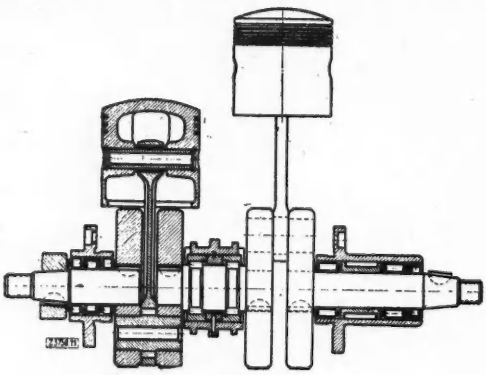


Fig. 6

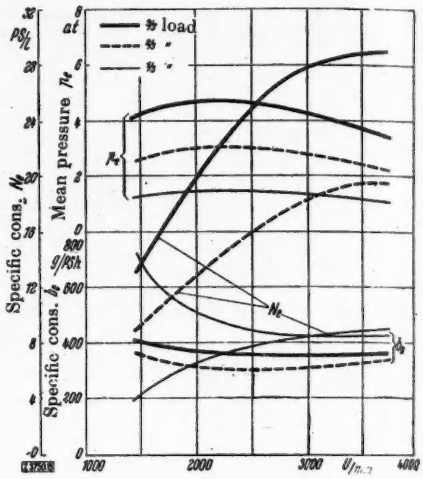


Fig. 7

To reduce the axial play of the shaft retaining rings are used. A shaft and bearings of the above type is illustrated in Fig. 6.

Fit and play of the bearings should be well designed to give smooth running of the shaft, and also for successful interchangeability. Generally the radial play of the bearing is from 0 to 15 μ, and only when this exceeds 30 μ can noises be detected with a stethoscope. The axial play of the bearings may be of the order of 100 to 200 μ. Assembled bearings are expensive, and, therefore, not in use. The chief advantage of roller bearings is that no special precautions need to be taken when starting the engine in cold weather.

Fuel Consumption and Output.

During the last few years the specific consumption of two stroke engines has been considerably reduced without altering the speed. The maximum speed of two stroke engines, at present, is about 4000 r.p.m., while many four stroke engines have speeds between 4000 and 5000 r.p.m. The maximum mean pressure of the latest two stroke engines is about 4.5 atm., and the mean pressure curve is approaching that of four stroke engines.

Fig. 7 shows the characteristic curves for a modern two stroke engine of the crank case compression type. The output is about 29-30 h.p./litre, and the fuel consumption is 300-350 gm./h.p./hr. The fuel consumption at partial

load is less than that at full load: this feature accounts for the very economical running costs of two stroke engines.

Lubrication.

The lubrication system, type of shaft bedding, and scavenge pump employed are in a very close relation to each other. Where separate scavenge pumps are provided a free choice of lubrication system exists. In this case roller bearings or gun metal bearings may be used with equally good results. On the other hand, if charging of the cylinders is by means of crank case compression, surface bearings cannot be used since the lubricant may leave the bearings and the degree of lubrication would be uncontrolled. Hence this type of scavenging necessitates mixed lubrication and roller bearings.

Although this type of lubrication has several disadvantages such as relatively high oil consumption, uniform and equal lubrication of all parts irrespective of special conditions and requirements, these are negligible compared with its advantages, which may be summarised as follows:

(1) Elimination of the lubrication oil pump and tubing.

(2) Elimination of failure of lubrication system.

(3) No dust can get into the bearings through the oil.

(4) All moving parts are lubricated.

(5) The lubricant does not deteriorate.

Owing to the last mentioned point the oil consumption with this type of lubrication is hardly higher than that of pressure lubrication. The oil fuel ratio, too, has been considerably improved during the last few years, by reducing it from 1:10 to about 1:25. Since the fuel consumption has been also reduced to about 300 gm./h.p./hr., the total reduction of oil consumption amounts to about 50%.

The quality of the lubricant is, naturally, a point of considerable importance. Many years' experience has proved that mineral oils are much superior to others at the medium working pressures which exist with two stroke engines. Even 1% of vegetable oil in the lubricant may cause serious troubles. Therefore, great care has to be exercised by selecting a pure mineral oil for these types of engines. The proper shaping of the inlet and outlet ports, suitable arrangement of piston rings, sufficient axial play, and dimensions of the piston are the main design factors influencing the lubrication.

Where separate scavenge pumps are used a special type of lubrication has been developed recently, which is a consumption lubrication, an intermediate between mixed lubrication and circulating lubrication. A pump rotating at slow speed supplies oil to the various bearing surfaces through grooves and holes in the shaft, etc. The oil is not returned to the pump, but it still operates on a very economical basis, and the oil consumption is about 3 to 4 kg/h.p./hr., i.e., about 1.0 to 1.2 litres/1000 km. The proper arrangement of grooves and channels requires considerable practical experience to avoid oil deposits and circulation failures in the lubrication system.

Scavenging by Crank Case Compression.

The only difficulty which might arise with this system, the efficient sealing of the crank shaft bearings, may be regarded as completely solved; collar rings, fully enclosed in sheet metal, are used, such that at the inside where sealing is required they fit closely to the crank shaft while at other parts they are only pressing lightly against the shafting. The collar rings are made of leather, rubber, or other plastic material, and they should be provided with an ample oil supply. The packing is able to withstand temperatures up to about 200° C., and is not affected by oil, petrol, or water.

Theoretically, the volume V_o, i.e., the volume in the crank case below the piston when this is at the O.D.C., determines the efficiency of the scavenging. Actually, however, the ratio V_2/V_3 is a more important factor (Fig. 8). The last part of the pressure rise up to p_x can be attributed to the back pressure into the crank case from the exhaust gases. The statical compression ratio, V_2/V_0, depends on the stroke/bore ratio (s/D). With values of s/D = 0.9 to

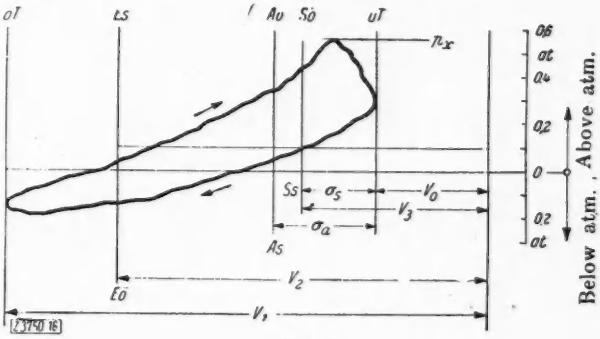


Fig. 8

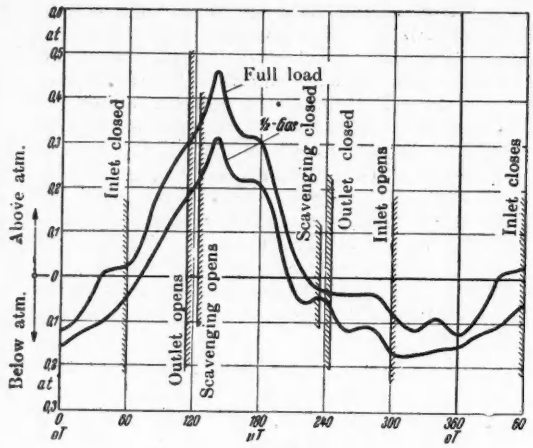


Fig. 9

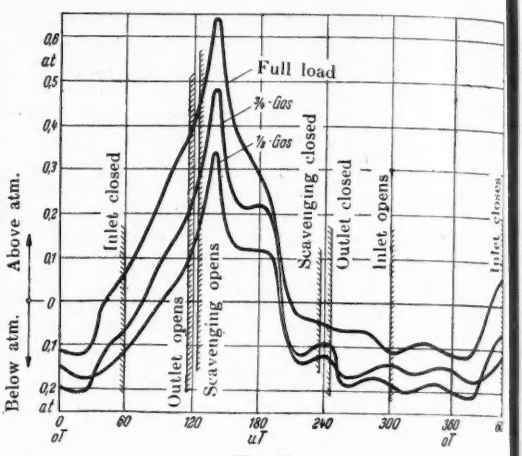


Fig. 11

1.1 δ becomes 1.8 to 1.3 or $\beta(V_o/V_h)=1.70$ to 2.50 (where V_h is the crank case volume below the piston), for stroke volumes of about 250 to 350 cc. The most economical charging is obtained if δ is about 1.40. As a rule δ should be at least as large as the exponent in the adiabetic expansion, i.e., at least 1.35.

These relationships are illustrated in Figs. 9-11. With too small a crank case volume the initial scavenging pressure would rise too rapidly; the consumption increases more rapidly than the power of the engine. Fig. 12 shows the efficiency of the crank case pumping with fully opened throttle as a function of speed. The improvements obtained by increasing δ to higher values than 1.40 are not considerable.

Large values of β, e.g. 3.0, have much less effect on the power and consumption as would be expected. Therefore, the ratio s/D may be as high as 1.25. However, the acceleration of engines with a large crank case volume is inferior.

Separate Scavenge Pumps.

The disadvantages of engines using separate scavenge pumps are the complicated crank shaft design and inferior balancing. Also it is very difficult to arrange the scavenging channels from the cylinder in a satisfactory way from a hydrodynamic point of view. The pumps are usually designed with a large plunger diameter and small stroke, so that the stroke/bore ratio is

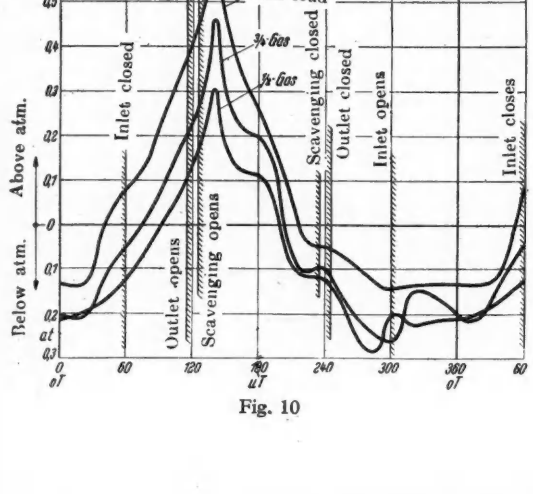


Fig. 10

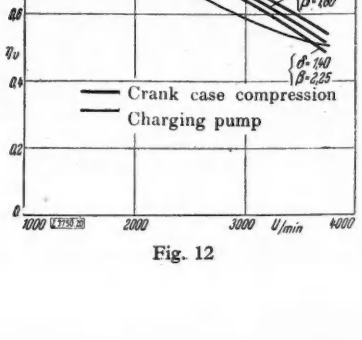


Fig. 12

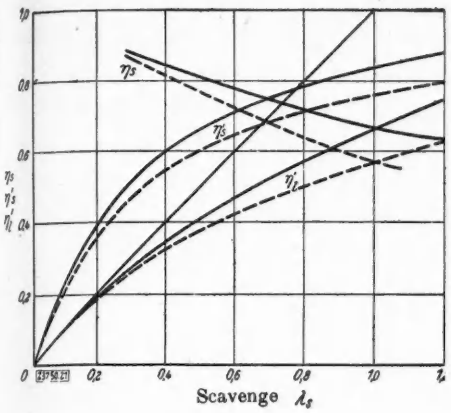


Fig. 13

0.6 to 0.7. Fig. 13 illustrates that the scavenging efficiency is considerably reduced in an engine governed by piston and ports. The upper limit of scavenge supply for these engines is, therefore, only about 0.9.

Scavenging.

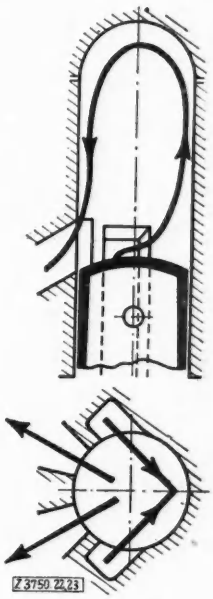
Efficient scavenging with small wastage of fuel is one of the most important factors in the two stroke engine design. The quantitative efficiency of scavenging becomes more important where the scavenge consists of a mixture of air-fuel, and is less important where the fuel is injected into the cylinders. Since two stroke engines are governed by the piston which covers and uncovers the ports, the speed has a considerable influence on scavenging. From this it follows that the scavenging cannot be arranged to be efficient over the entire speed range. Further development is expected to overcome this difficulty by using slide valves to alter the cross section of the ports at varying speeds.

In the most recent types of engine the scavenging ports are so arranged that the scavenge enters the cylinder above the piston at an angle such that it is blown against the cylinder walls facing the outlet ports (see Figs. 14 and 15). The scavenging pressure varies with speed. The pressure variations are of such nature that the theoretical prediction of the scavenging process is impossible. It might, however, be stated that with the above type of scavenging ports the flow is controlled by the cylinder walls; turbulence is reduced to a minimum, with the result that the burnt products do not mix with the scavenge, but are driven out in front of the fresh charge. A well designed scavenging system ensures uniform expansion of the cylinder walls by avoiding hot and cold spots on it: a large surface area of the cylinder should be swept by the scavenge.

Combustion.

For most engines the scavenging is most efficient at speeds of about 2000 r.p.m. Since, at this speed knocking starts in the engine cylinder when the throttle is suddenly opened, it must be concluded that lack of the residual charge is responsible for the knocking. Therefore, it would be essential to design a combustion chamber which would be capable of quick heat transfer without greatly influencing the scavenging. The problem could also be solved by using fuel of a high anti-knock value.

It was found that at high speeds and throttled fuel supply pre-ignition may take place. This is probably due to the presence of a large proportion of the hot residue gases which causes pre-ignition at various nudei and causes pulsations in the gases. This theory is supported by the increased speed of combustion, instability, and distinct dependence of speed. It remains to be shown whether this is principally a scavenging problem or one of thermo-dynamics.



Figs. 14 and 15

Cooling.

Practically all two stroke engines for motor cars are designed with water cooling, while those for motor cycles employ air cooling. Future designs are expected to extend air cooling to motor car engines.

Piston slap has been considerably decreased in later designs which make use of high quality aluminium alloys with high silica content for the piston.

INVESTIGATION OF THE INFLUENCE OF BENDING AND BUCKLING ON STRESS MEASUREMENTS IN MONOCOGUE STRUCTURES ACCESSIBLE FROM OUTSIDE ONLY

By A. Dose, Focke-Wulf Flugzeugbau G.m.b.H., Bremen. (From *Luftfahrtforschung*, Vol. 18, Nos. 2/3, March, 1941, pp. 95-101).

When determining stresses in thin sheet structures by strain measurements the results may be appreciably altered owing to the fact that after buckling the sheet some undesirable strain caused by bending comes into play. In order to get true results this should not be neglected, even if the folds or wrinklings are not visible by the eye. The unequal elongations of the inner and outer side of the buckled sheet can be eliminated by taking strain measurements on both sides. If however, as in many modern aircraft structures, there is no access from both sides, other ways have to be developed to obtain true mid-sheet strains. The magnitude of the strain due to bending can be derived by the help of Fig. 1. A sheet of thickness (d) is bent

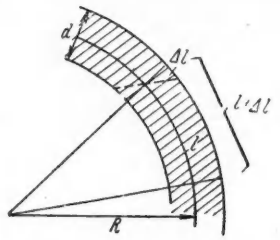


Fig. 1

so that the mid-sheet with a radius of curvature R does not experience any change of its original length. The strain on the outer surface is denoted by e. Then from Fig. 1 it follows, that

$$(1)\quad \frac{l}{R}=\frac{l+\Delta l}{R+{}^{d}/_{2}},\ \text{or}\ \frac{\Delta l}{l}=e=\frac{d}{2}\cdot\frac{1}{R}$$

whereby linear strain distribution was assumed.

According to the basic requirement that all measurements should be made from the outside of the structure only, the curvature of the sheet can be measured by a simple instrument such as shown in Fig. 2 which is adapted from the spherometer employed in lens measurements and incorporated in the standard Huggenberg Extensometer.

The projection of the curved surface can be assumed with reasonable accuracy to be a circle and then (s) represents a chord to it. The pointer will indicate on the scale the displacement (f) such that

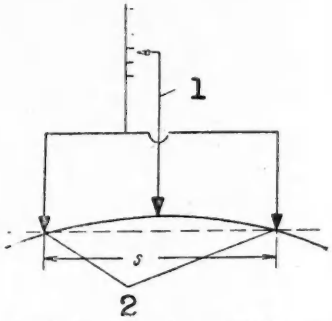


Fig. 2 (1) Movable pointer. (2) Fixed points.

$$(r-f)^2+\left(\frac{s}{2}\right)^2=r^2,$$

$$(2)\quad r=\frac{f^2+({}^{s}/_{2})^2}{2f}=R+\frac{d}{2}$$

If (f) is small, we can write

$$(3)\quad R=\frac{({}^{s}/_{2})^2}{2f}-\frac{d}{2}$$

and the combination of equation (1) and (2) gives

$$(4)\quad e=\frac{d\,.\,f}{f^2+({}^{s}/_{2})^2-df}$$

Neglecting f^2 in equation (4) we get

$$(5)\quad e=\frac{d\,.\,f}{({}^{s}/_{2})^2}\cdot\frac{1}{1-\frac{df}{({}^{s}/_{2})^2}}$$

If $\frac{df}{({}^{s}/_{2})^2} < 1$ then equation (5) can be expanded by the series

$$e=\frac{df}{({}^{s}/_{2})^2}\left[1+\frac{df}{({}^{s}/_{2})^2}+\ldots\right]$$

Thus approximately

$$(6)\quad e\doteq\frac{df}{({}^{s}/_{2})^2}$$

In this equation f is positive for convex curvature (see Fig. 3). Under normal conditions the error evolved in equation (6) is very small, as shown in the following table.

TABLE I.

(d = 1 mm., $s/2$ = 10 mm.).

f mm.	e.10^4 from equ. (1) & (2)	e.10^4 from equation (6)	e.10^4
0.1	10.01	10.0	0.01
0.01	1.0001	1.0	0.0001
0.001	0.100001	0.1	0.000001

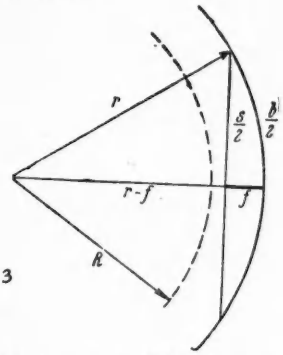


Fig. 3

If the displacement of the sheet under load changes from a certain value f to the value f_2 (see Fig. 4) the increase in the strain due to bending amounts to

$$(7) \qquad e = \frac{d}{(s/2)^2}\,(f_2 - f_1)$$

$$(8) \qquad = \frac{d}{(s/2)^2}\,\Delta f$$

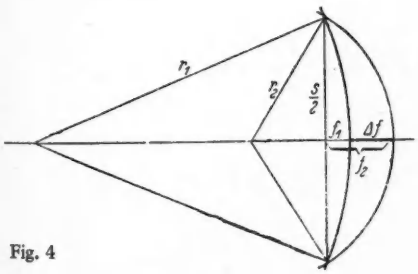


Fig. 4

An extensometer, the instrument shown in Fig. 2 can be easily combined. In this case allowance has to be made for the elongation of the material under test after the two arms of the instrument are fixed to the spherical surface. Instead of the constant distance (s) we get (s_1) and (s_2) respectively, and the exact form of the strain equation becomes

$$e = d\left[\frac{f^2}{(S_2/2)^2} - \frac{f_1}{(S_1/2)^2}\right]$$

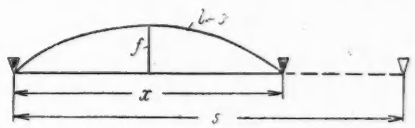
As the difference between (s_1) and (s_2) is negligible in most cases, equation (8) can still be used. In two worked out examples it was shown that the error made by using equation (8) is about 0.2%.

In Fig. 5 an extensometer is fixed on to a flat sheet, the gauge length being (s). The mid sheet experiences now a deformation without any change in its length, so that (b) the length of the deformed mid-sheet remains equal to (s). The extensometer, however, indicates a change of the distance (s) to (x) due to the displacement (f). As the extensometer measures the change in length of the chord, it is obvious that for b=s an apparent strain (e) is produced.

$$(9) \qquad \text{Hence} \quad c = \frac{s-x}{s}$$

From Fig. 5 it follows

$$(10)\ \sin\left[\frac{180}{\pi}\cdot\frac{s}{2}\cdot\frac{2f}{f^2+(x/2)^2}\right]^{\circ} = \frac{x}{2}\cdot\frac{2f}{f^2+\left(\frac{x}{2}\right)^2}$$



▽ Starting position of the extensometer.
▼ Position after deformation.

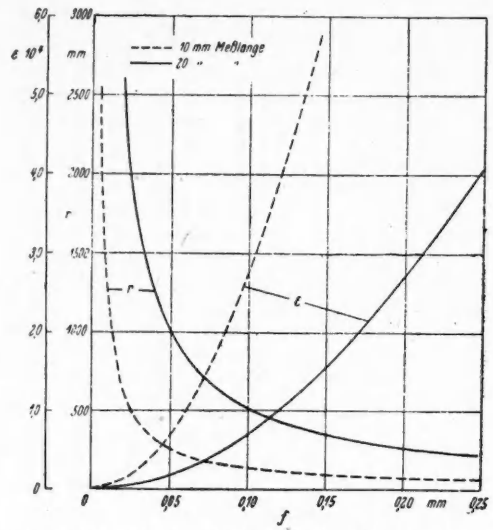
Fig. 5

The combination of equations (9) and (10) yields approximately a parabola, the equation of which is

$$(11) \qquad e = \frac{2}{3}\cdot\frac{f^2}{(s/2)^2}$$

In Fig. 6 the apparent strain from equation (11) and (r) from equation (2) are plotted against the displacement (f). The apparent strain may be of considerable magnitude, and therefore, a correction $\Delta e = e_2 - e_1$ should be applied according to $\Delta f = f_2 - f_1$ in equation (8).

The correct value of (e) in equation (8) depends also on the correct measurement of (f). In the derivation of the formulas above the points of contact between gauge and sheet were assumed to be on the sheet surface. Actually the influence of the depth of penetration of the



(10 mm Messlänge) s = 10; (20 mm Messlänge) s = 20.

Fig. 6

points into the sheet should be accounted for. The thinner the sheet the greater the influence. However, in the present case this has been neglected. Another assumption was the constant curvature of the sheet between the gauge points. In the following the influence of a variable curvature on equation (8) will be analized. For this purpose a strip of originally flat sheet d = 1 mm. will be bent so that the mid sheet, without any change in its original length, satisfies the equation

$$(12) \qquad y = A \sin\left(2\pi \frac{x}{\lambda}\right)$$

with A=1.5 mm., and λ 100 mm. (see Fig. 7).

Two methods were employed :

(1) Exact solution by integration.

(2) The " Three-point " method.

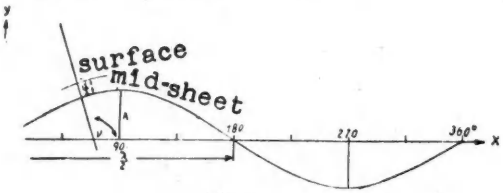


Fig. 7

In Fig. 8 various ratios of curvature are shown for the curve y=sin x ; the radius of curvature being ρ. To reduce operations,

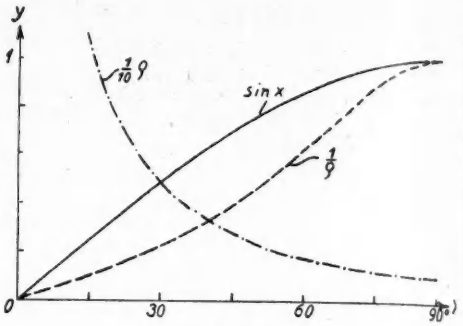


Fig. 8

calculation for one point of the sine curve has been made only. This is where the greatest change in $1/\rho$ is encountered and from Fig. 8 it can be seen that the chosen point comes near 65°.

The Integral Method.

The strain at the surface can be easily derived (see Fig. 9)

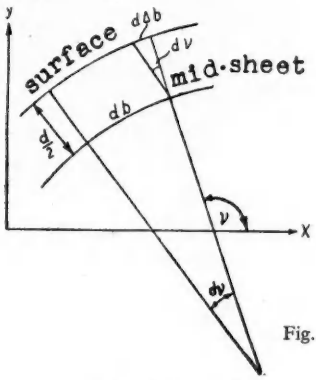


Fig. 9

$$(13) \qquad \frac{d \cdot \Delta b}{dv} = \frac{d}{2}$$

The total elongation within the limits 1 and 2 is then

$$(14) \qquad \Delta b = \frac{d}{2} \int_{v_1}^{v_2} dv$$

With y from the equation (2), the angle v can be given as

$$\tan v = -\frac{1}{y^1} = -\frac{1}{A \frac{2\pi}{\lambda} \cos\left(\frac{2\pi}{\lambda} \cdot x\right)}$$

Thus

$$\frac{1}{\cos^2 v} \cdot dv = - \frac{\frac{2\pi}{\lambda} \sin\left(\frac{2\pi}{\lambda} \cdot x\right)}{\Delta \frac{2\pi}{\lambda} \cos\left(\frac{2\pi}{\lambda} \cdot x\right)} dx$$

and

$$(15) \quad dv = - \frac{A\frac{4\pi^2}{\lambda^2} \sin\left(\frac{2\pi}{\lambda} \cdot x\right)}{1 + A^2 \frac{4\pi^2}{\lambda^2} \cos^2\left(\frac{2\pi}{\lambda} \cdot x\right)} \cdot dx$$

Consequently from equation (14)

$$(16) \quad \Delta b = - \frac{d}{2} \cdot \frac{A 4\pi^2}{\lambda^2} \int_{x_1}^{x_2} \frac{\sin\left(\frac{2\pi}{\lambda} \cdot x\right)}{1 + A^2 \frac{4\pi^2}{\lambda^2} \cos^2\left(\frac{2\pi}{\lambda} \cdot x\right)} \cdot dx$$

By means of transformation

$$(17) \quad z = A \frac{2\pi}{\lambda} \cos\left(\frac{2\pi}{\lambda} \cdot x\right)$$

$$dz = - A \frac{4\pi^2}{\lambda^2} \sin\left(\frac{2\pi}{\lambda} \cdot x\right) dx$$

we get

$$(18) \quad \Delta b = \frac{d}{2} \int_{z_1}^{z_2} \frac{1}{1+z^2} \cdot dz = \left[\frac{d}{2} \tan^{-1} z\right]_{z_1}^{z_2}$$

Finally by substituting equation (17) in equation (18)

$$(19) \quad \Delta b = \left\{\frac{d}{2} \tan^{-1}\left[A \frac{2\pi}{\lambda} \cos\left(\frac{2\pi}{\lambda} \cdot x\right)\right]\right\}_{x_1}^{x_2}$$

The strain is

$$(20) \quad e_{\text{integr}} = \frac{\Delta b}{b}$$

and (b) from the sine curve amounts to

$$(21) \quad b = \int_{x_1}^{x_2} \sqrt{1+y^2} \cdot dx =$$

$$\int_{x_1}^{x_2} \sqrt{1 + A^2 \frac{4\pi^2}{\lambda^2} \cos^2\left(\frac{2\pi}{\lambda} \cdot x\right)} \cdot dx$$

Hence with this method the strain is defined by equations (19) (20) and (21).

The Three Point Method.

Consider arch (b) of the mid-sheet between the points P_1 and P_2. The co-ordinates of these points are (x_1, y_1) and (x_2, y_2) respectively (see Fig. 10). The point (x_m, y_m) halves the chord

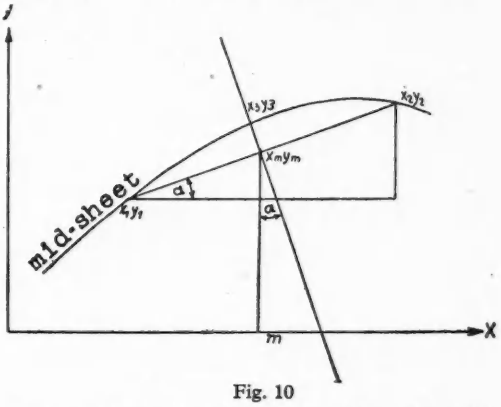


Fig. 10

between (x_1, y_1) and (x_2, y_2). Now the third point P_3 (x_3, y_3) can be found as the intersect on of the sine curve with the perpendicular to the chord at (x_m, y_m). The equation of the perpendicular is

$$(22) \quad x = - \frac{m}{y_m} \cdot y + x_m + m$$

therein

$$(23) \quad \frac{m}{y_m} = \frac{y_2 - y_1}{x_2 - x_1}$$

By substituting equation (22) in equation (21) and solving the transcendental equation by approximation, y_3 is obtained, the corresponding x_3 to which is found from equation (22). The points P_1, P_2 and P_3 should now define a circle, the radius R of which is to be found. For the co-ordinates (x_m, y_m) of the centre of this circle the following equation can be written:

$$(24) \quad (x_n - x_m)^2 + (y_n - y_m)^2 = R^2 ; \; n = 1, 2, 3, \ldots$$

Substituting R in equation (1), $e_{3\text{-point}}$ is obtained.

As mentioned before a calculation has been made for the sine curve of the mid-sheet at 65°. The difference between the two methods has been determined:

$$e_{3\text{-point}} - e_{\text{integr.}} = \Delta e$$

where R had values between 17.04 and 46.4 cm. Two gauges have been used with the gauge points s=10 mm., and s=20 mm. distant; the result was

$$\Delta e_{20}=3.4\% \text{ and } \Delta e_{10}=0.8\%$$

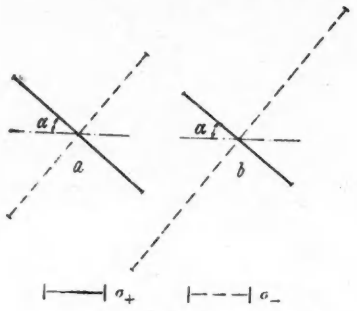


Fig. 11.

Measurements and calculations have shown that gauges with s=20 mm. were satisfactorily accurate in most of the practical cases. Measurements on a light alloy spar web d=0.75 mm. show how the stress distribution is affected by strain due to bending in the fold of the web. Fig. 11 gives a picture of the direct stresses σ_1 and σ_2 for

(a) corrected e-values according to the present investigation

(b) for uncorrected e-values.

The values are tabulated in Table II., comprising the maximum shear stress τ_{max} and angle α.

TABLE II.

	a	b
σ_1	−678 kg/cm²	−1060 kg/cm²
σ_2	+558 kg/cm²	+460 kg/cm²
τ_{max}	618 kg/cm²	760 kg/cm²
α	−37.5°	−36.4°

MEASUREMENT OF THE PERIOD OF NATURAL VIBRATION OF AN AIR-SCREW BLADE

By JUICHI OBATA and YAHEI YOSIDA. (From *The Aeronautical Research Institution*, Tokyo Imperial University, Vol. 15, No. 191, July, 1940, pp. 99-108).

A NUMBER of methods have been already devised for determining the natural frequency of the non-rotating airscrew blade and for establishing the nodal lines of the blade. In the following a method will be described which is claimed by the authors to be a very simple one and frequencies of various modes of vibration, bending as well as torsional, can be easily determined by this method up to those of very high order.

The method consists of the following. An electrical condenser is formed between the surface of the vibrating body and an electrode which is a slightly modified micrometer head of the usual kind; the former being in the present case an airscrew blade firmly clamped at its boss to a stout stand of concrete. The condenser or capacity is introduced into an electrical circuit, and the change in it caused by the vibration is determined electrically. Fig. 1 shows the electrode that forms an electrical condenser with the vibrating body. Compared with previously adopted constructions two improvements were made.

(a) The micrometer head in the arrangement shown in Fig. 1 rotates when the width of the air gap is adjusted. Owing to the air gap being often very narrow, a small fraction of a millimeter, some inaccuracy is caused when the surface of the vibrating body is not sufficiently flat, as in the case of an airscrew blade. A somewhat complicated arrangement was therefore adopted; the micrometer head was so constructed that it would not rotate when the micrometer screw was turned.

(b) The second improvement is electrical. The vibrating body is usually connected to earth. If the electrode, which is placed against the earthed vibrating body is exposed, as in previous arrangements, owing to electromagnetic

(1) Air gap 0.3 mm.
(2) Airscrew blade.

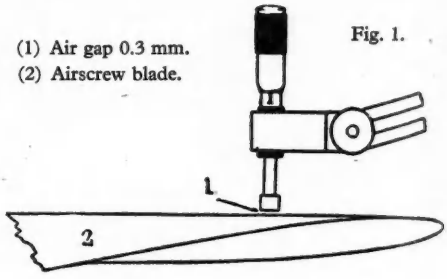


Fig. 1.

(M) Micrometer
(S) Support
(B) Insulator
(G) Guard-ring
(E) Electrode

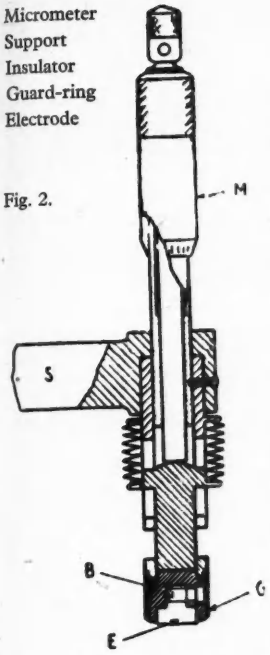


Fig. 2.

induction at the condenser part, a 50 cycle alternating current of the lamp circuit, or similar electrical disturbances, an error in the final indication of the vibration results. In order to overcome this difficulty, an insulated guard ring of metal was inserted around the electrode, with very satisfactory results. The final construction, with these two improvements introduced, is shown in Fig. 2. Fig. 3 shows the diagram of the electrical circuit.

Records of vibration can thus be obtained by taking oscillograms of the output of the amplifier, after giving a single blow to the body, the vibration of which is to be determined. In the case of a vibrating body, such as an airscrew blade, the vibration is very complicated, several different modes of vibration are excited at the same time, making the oscillogram so complex that it is not easy to obtain from it any accurate determination of the frequencies of the various modes, both flexural and torsional.

This difficulty is, however, easily avoided by inserting a set of suitable electrical filters in the circuit. In the present experiments fairly satisfactory results could be obtained by using only two filters, namely a high-pass (cutting frequency 150 cycle) and a low-pass (cutting frequency 200 cycle) filters. Obviously, better results can be obtained by using a number of band-pass filters.

In Fig. 4 are produced some examples of oscillograms thus obtained, using filters. As the timing wave, a 1000 cycle a.c. wave was recorded together with the vibration. In every case the airscrew blade was set in vibration by striking it only once at a point on the trailing edge near the boss, the vibration being determined at a point near the blade tip.

Whether the recorded vibration is bending or torsional can easily be decided by comparing two records taken at two points, one on the edge and the other on the central line of the blade. Torsional vibration can only be recorded at the edge, whereas flexural vibration can be recorded on both points—on the edge as well as on the central line. It will be seen in Fig. 4 (a) that the lowest mode of the bending vibration (frequency about 30) and the second mode of the torsional vibration (frequency about 635) were remaining for a long period of time. The positions of the nodes and the antinodes of vibration can also be easily determined by comparing the vibrations of several points keeping the position of the striking point constant and changing the measuring point, the air gap between the electrode and the vibrating body being kept as constant as possible.

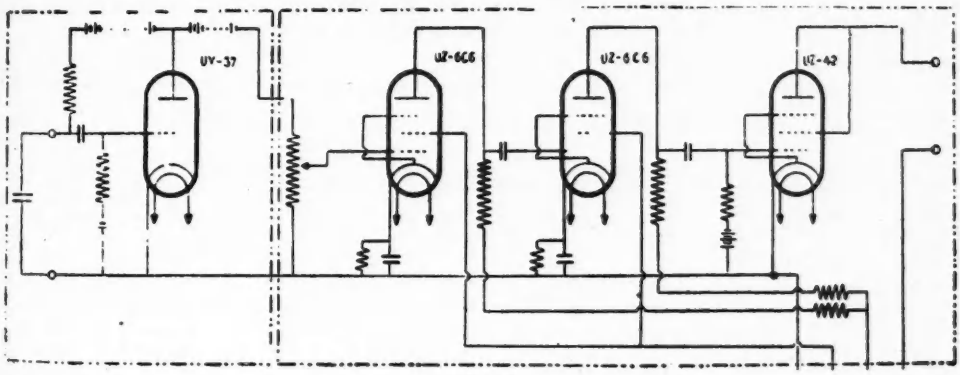


Fig. 3

The authors, who further attempted to use an electrical frequency analyser instead of the oscillograph, found that the former apparatus gives very satisfactory results in vibration measurement. In the usual practice of applying harmonic analysis to the wave form oscillogram, analysis is evidently limited to that case in which the components are harmonic, whereas in the electrical frequency analyser, it does not matter whether the components are harmonic or not, an advantage of the new procedure which makes it very suitable for analysing vibrations such as those of an airscrew blade. But since in the frequency analyser at least 30 to 40 seconds are required for a single analysis, it is necessary to repeat the blows during this time. For this purpose, the blow may be given by hand, but it is more convenient to use some mechanical or electrical means.

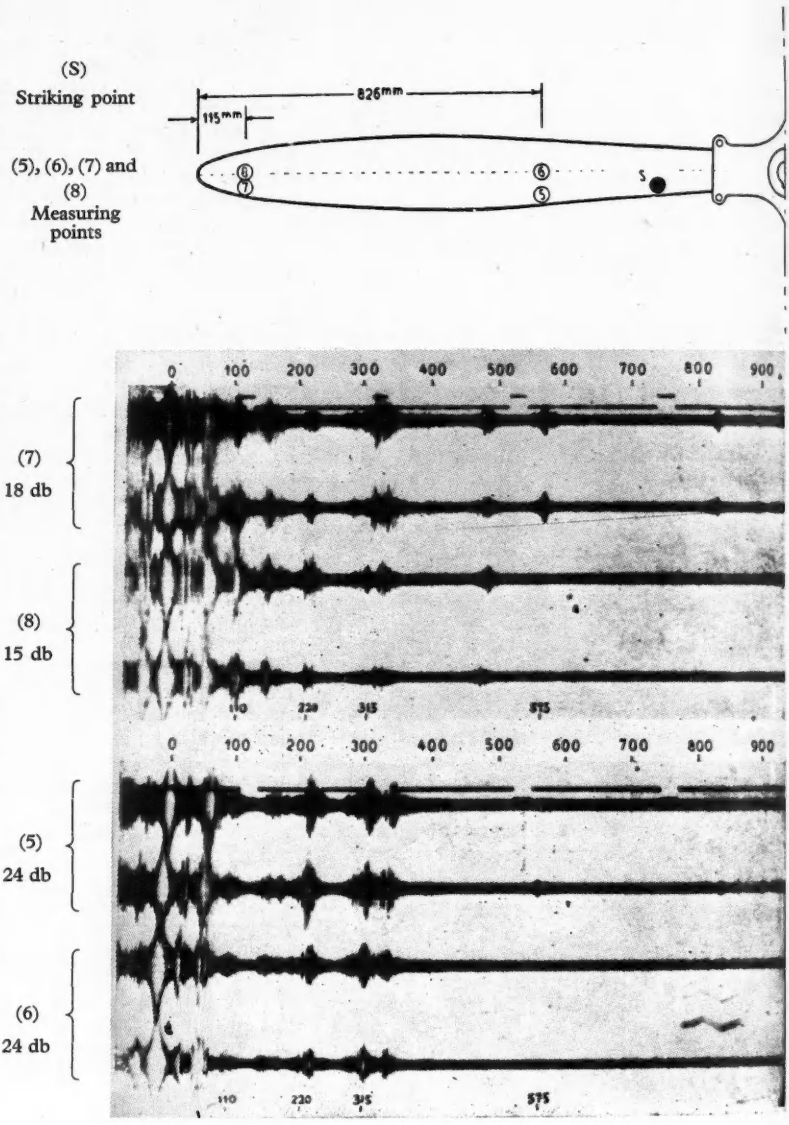
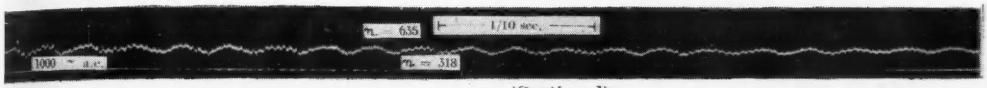


Fig. 5 Records of Frequency Analysis of the Vibration.

By using an electrical frequency analyser, several vibration analyses of airscrew blade were made, some of the records being reproduced in Fig. 5. An inspection of the graphs readily shows that, by means of a single analysis, the frequencies of various modes extending over a very wide frequency range, 30 to 1000 and more, as well as the manner of damping of the vibration, can be clearly presented, and by

Fig. 4 (a) Record, taken without filter.



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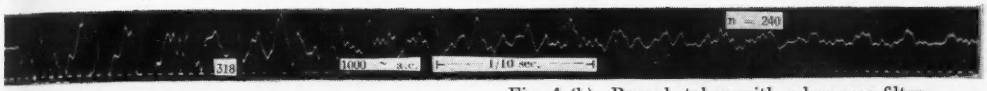


Fig. 4 (b) Record, taken with a low-pass filter.

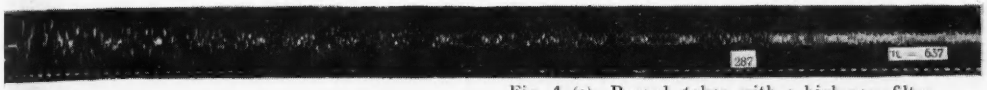
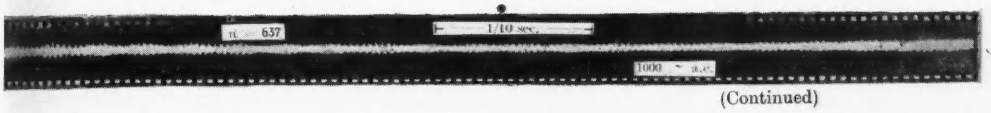


Fig. 4 (c) Record, taken with a high-pass filter.

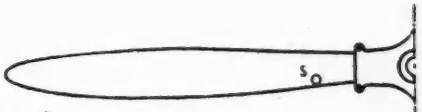


(Continued)

Fig. 4

comparing two records of analyses it is easily decided whether the recorded component is bending or torsional vibration.

The scale of the abscissa of the record, that is the frequency scale is, however, fairly rough and somewhat liable to change, so that very high accuracy in the value of the frequency cannot be expected. It is, however, not difficult to guarantee the value to a few per cent or less,



(S) Striking point.
(1) Leading edge; (2) Central line; (3) Trailing edge. Abscissa: Distance from the tip; Ordinate: Amplitude of vibration in arbitrary scale.

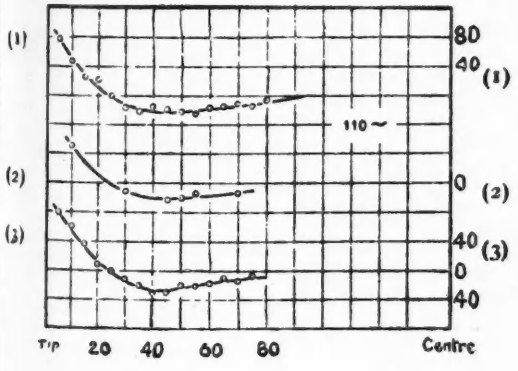


Fig. 6

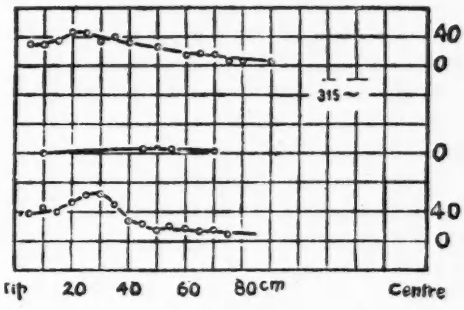


Fig. 7

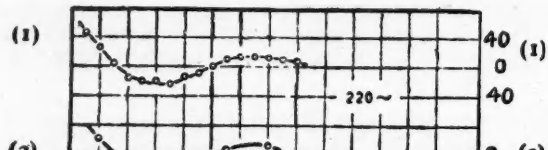


Fig. 8

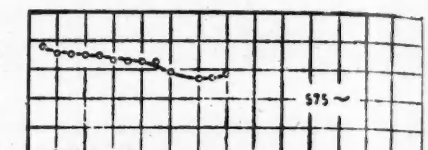


Fig. 9

provided the scale is frequently calibrated with the aid of a set of tuning forks, or some such appropriate standards.

Figs. 6-9 show a typical example of the various modes of vibration, flexural and torsional, of an airscrew blade, obtained by means of records secured with the frequency analyser.

Obviously, the application of the present method is not restricted in the case of an airscrew blade, it may be applied to various vibrating bodies, metallic as well as non-metallic. In the latter case it is sufficient by making it conductive by a strip of thin tin-foil pasted on its surface.

APPARATUS FOR MECHANICAL TESTING OF METALS

By D. M. FANCHENKO. (From *Zavodskaja Laboratoria*, Moscow, No. 2, 1941, pp. 131-133).

THE Russian State Testing Machine Works (G.Z.I.P.) is now producing a total of 16 types of appliances for the mechanical testing of metals. The following details of the latest types will be of interest.

Fig. 1 shows a portable Brinell tester PB3. It consists essentially of a springy horseshoe frame of heat-treated alloy steel. One arm of the frame carries the specimen-block mounted on an adjusting screw, while the other takes the testing head with its hardened ball. When the object to be tested is placed between the block and the testing head, and the handle is rotated, the ball presses into the metal while the load which causes it to do so sets up a deformation in the frame which is recorded on the dial of an

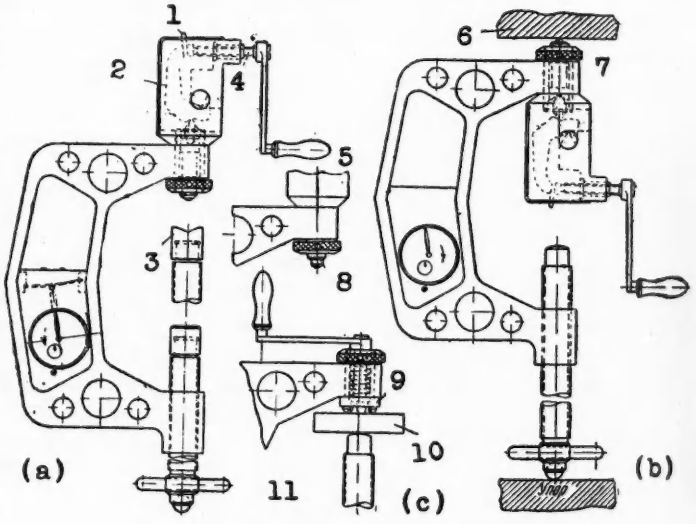


Fig. 1. Portable Brinell Press.
(1) Indicator; (2) Head; (3) Block for circular specimens; (4) Loading screw; (5) Head; (6) Metal to be tested; (7) Head reversed; (8) Mounting for 5 mm. ball; (9) Milling cutter; (10) Specimen; (11) Surface cleaning attachment.

indicator mounted within the frame. A calibration table is supplied showing indicator readings at the standard loads of 3000, 1000 and 750 kgs., and the handle is rotated until the necessary standard loading is attained. The load is applied for the regulation period of time and on its removal the size of the impression is measured with the aid of a microscope supplied with the machine.

An alternative method of use is indicated at (b), while at (c) is shown an attachment for cleaning the surface of the metal to be tested.

The weight of this portable Brinell press is 37 lbs., maximum net span between ball and support 7 ins., and maximum distance from C. L. of ball to the body of the frame 3½ ins.

These portable Brinell testers are used mainly for testing large castings, forgings and machine parts, they can also be used for small articles, but for these the standard Brinell machine is more convenient.

The G.Z.I.P. Works also make two standard types of Brinell machines: the "Laboratory" (LB) and the "Workshop" (RPB). The Workshop type is intended, in addition to ordinary tests, to meet the need for rapid testing of a large number of mass-produced articles, and is capable of dealing with as many as 650 specimens per hour. Being for workshop use it is of robust construction, is controlled by means of a foot-pedal, and equipped with an indicator showing the size of impression.

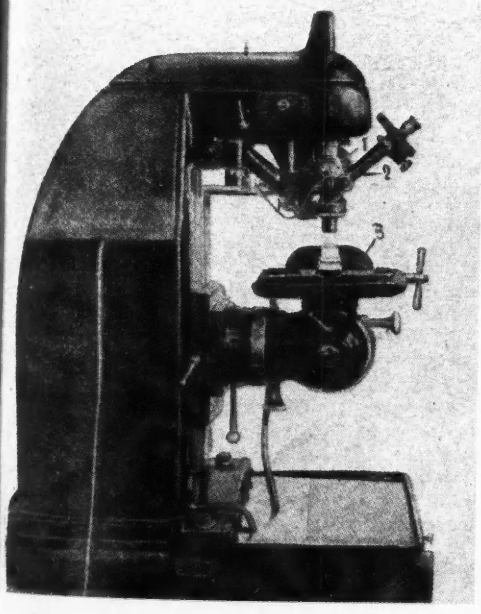


Fig. 2. Wear Testing Machine.
(1) Disc of hard steel; (2) Microscope; (3) Movable table.

Both the LB and the RPB types are electro-motor operated and are partially automatic: once the motor is switched on the correct load is applied automatically, and the motor is automatically switched off after a pre-determined period.

Tensile testing machines (also electro-motor operated) are made in 5-ton and 1-ton sizes, the former being reversible, i.e., suitable for compression tests. There are likewise two sizes of pendulum impact testing machines (15 and 30 kg. hammers), a reversible bending tester (type "G") and an Eriksen press for sheet metal tests (type "E").

Fig. 2 shows a wear testing machine (type "I"), introduced this year. The wear is produced by a disc of hard alloy steel 30 mm. diam. by 2.5 mm. width, rotating at approximately 700 r.p.m., and pressed down on to the surface to be tested by a load of 15 kgs. The wear-resisting quality of the surface is measured by the volume of material removed in 3000 turns of the testing disc, so that the test takes about 5 minutes. The length of the resulting groove is then measured with the aid of a microscope, mounted on the machine itself. As will be seen from Fig. 2 this testing machine has a heavy frame and incorporates several movements and attachments. The load is applied by a system of levers. The rubbing disc is electrically driven and cooling of the surface is taken care of by a circulating pump. The table is adjustable, to permit of testing articles of varying shapes and sizes. The machine is intended particularly for testing the quality of the skin of the metal parts.

Fig. 3 shows the fatigue testing machine. This takes a standard specimen of 7.52 mm. diam., with cylindrical head. The load is applied at the centre and the machine runs at 2,850 r.p.m. A circulating pump is incorporated.

For checking testing machines the G.Z.I.P. Works produces "dynamometers" as follows: 200 kg type DK 02 for Rockwell and Vickers machines, 3 ton type DK 3 for Brinell presses, and 50 ton type D 50 for tensile testing machines. Dynamometers DK 02 and DK 3, first put into production last year, are of an ingenious con-

struction proposed by a senior designer of the Works, Rezik.

As the main causes of inaccuracy in dynamo meters are friction and backlash, the transmitting mechanism in the DK Dynamometers is replaced by a flexible plate. As will be seen from Fig. 4, the dynamometer consists essentially of a strong but flexible frame (1) which deforms under load. Between the inner surfaces of this is placed a spring-steel plate (3) which bends in accordance with the deformation of the frame and its movement is recorded by the indicator (2). DK dynamometers are primarily designed for compression loadings, but can also be used for tensile testing machines with the aid of the usual reversing device. The adoption of the transmitting device described above and of a frame-form arrived at after a number of experiments have given good results, both as regards permanency of callibration and in obtaining identical readings with increasing and decreasing loads.

The 50-ton dynamometer D 50, first put into production this year, is also based on the flexible frame principle, but is designed for tensile loads.

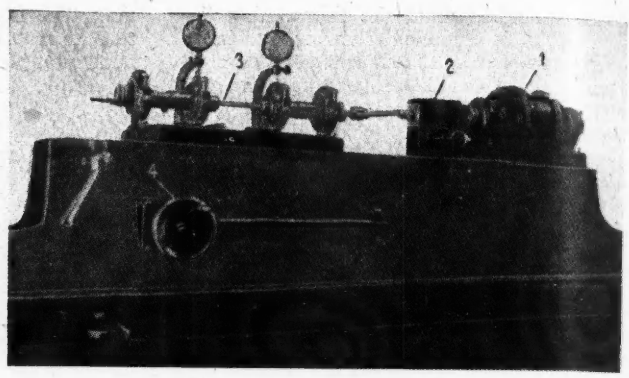
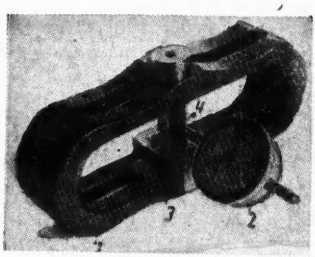


Fig. 3. Fatigue Testing Machine.
(1) Electrometer ; (2) Rev. counter ; (3) Specimen ; (4) Loading Flywheel.



(1) Flexible frame
(2) Indicator
(3) Spring-Steel Plate
(4) Indicator measuring road

Fig. 4. Dynamometer

RADIOGRAPHIC STUDY OF THE STRUCTURAL CHANGES OF STEEL AT HIGH TEMPERATURES

By P. N. Peredisty. (From *Zavodskaja Laboratoria*, Moscow, No. 2, 1941, pp. 158-61).

Radiographic study of the breaking down process of austenite has hitherto proved impracticable owing to the time of exposure required being too long to keep pace with the process to be studied. By focussing a broad beam, however, and making the camera aperture coincide with the focal point of the X-ray tube, this time has been recently reduced to as low as 10 to 60 secs.

This development has made it possible to devise a method by which the behaviour of super-cooled austenite could be studied in the second thermal region (i.e., above A t″) and which can be applied in other cases where the structural changes of an alloy are not too rapid for the requisite time of exposure.

The isothermal break-down of austenite was studied by recording the intensities of lines (111)γ and (110)α under Kα Cr rays. These lines were chosen because their good definition and small difference in angle of reflection. A special camera was constructed permitting of a temperature of specimen of 500 to 600° C., suitable for small angles of reflection.

Another set of experiments, with the object of measuring the parameters of lattice of austenite in the course of its breakdown, was carried out in a camera designed for large angles of reflection. In this case the line (113)γ was studied under Kβ Cr rays. The large angle of reflection helps to ensure accuracy. A great drawback is faintness of line in the Kβ rays, but it was hoped that this would be compensated by the absence of duplicating effect which is detrimental to accuracy of measurement. As the specimen was placed at a slight angle to the primary beam, only one-sided radiographs could be obtained. In view of the large-grained structure of super-

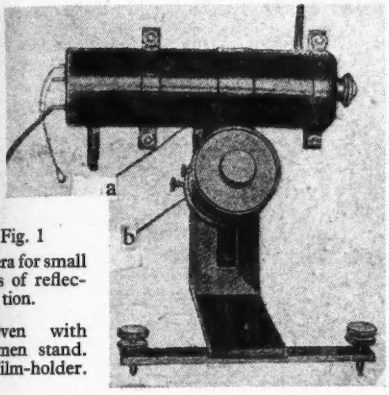


Fig. 1

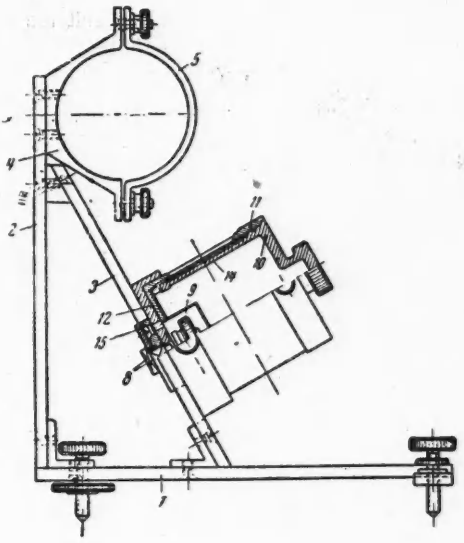
Camera for small angles of reflection.

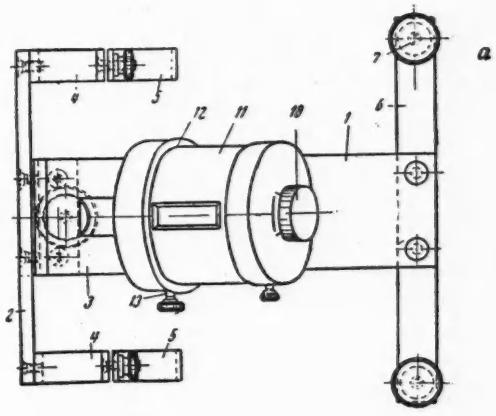
(a) Oven with specimen stand. (b) Film-holder.

cooled austenite, the apparatus was provided with a means of imparting an oscillatory movement to the film-holder.

Both the above cameras were arranged to take a series of 20 exposures on the same strip of film. In the first set of experiment the distance between film and specimen was 55 mms. and exposure time 1-2 minutes. In the second, the corresponding figures were 50 mms. and 1-3 mins.

(a) **Camera for Small Angles of Reflection** is shown in Figs. 1 and 2. The frame 1, 2, 3 in Fig. 2a carries the "oven" with its specimen stand (Fig. 1) and the film holder (b). The oven is secured by means of clamps (4 and 5), and the whole is adjustable to the X-ray beam by screws (6, 7). The film holder position, relative to the specimen, can be varied by means of the nut and screw (9, 8). The holder comprises a drum (10) and casing (11) fixed in a base-plate (12) by means of clamping screw (13).





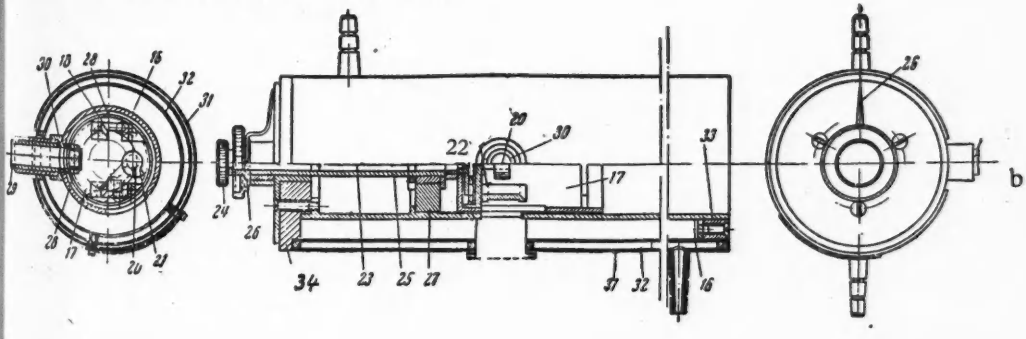


Fig. 2

Design of Camera for small angles of reflection. (a) Frame and Film holder ; (b) Oven and specimen stand.

The casing has a rectangular slot to admit the rays. The film, protected by black paper, is wound round the drum (10), and secured by a spring band (14).

The oven consists essentially of an iron tube (16), surrounded by a heating coil. Inside this tube is an iron cylinder (17), which carries the specimen stand and its securing mechanism. The stand is made up of two grooved projections (18) between which the specimen is introduced with the aid of a special carrier. The specimen is secured in position by the spiral-shaped clamp (20), mounted on the spindle (21), which has a projecting arm that is engaged by a radial slot in the disc (22), mounted on spindle (23), actuated by knob (24). Thus the specimen can be clamped by rotating knob (24).

Cylinder (17) is fixed to the hollow spindle (25) with a handle-cum-indicator (26). By rotating (26), therefore, the specimen can be turned to any desired angle. The entering of the cylinder in the body of the oven is effected by means of the plates (27, 28). The primary beam enters the camera through the collimator, comprising the tube (29) and the grid diaphragm, fitted into boss (30).

An aperture through which the reflected beam leaves the oven is located at an angle $2 = 66°$ to the axis of the collimator. The oven is surrounded by a water jacket (31, 32), secured by end plates (33, 34). Temperature is measured by means of a thermo-couple secured either to the specimen itself or to its stand. The focussing of lines (111)γ and (110)α is effected by adjusting the angle at which the primary beam strikes the specimen.

(b) **Camera for large Angles of Reflection,** shown in Fig. 3. The frame carries the oven and a flat film holder. The oven is secured by clamps which are movable along the guide bars, and can be secured once the correct position has been found, by means of the fixing screw.

At the top of the frame is the diaphragm which is also the axis of the oscillatory motion of the film holder. This motion is effected through the medium of a worm and wheel mechanism actuating disc which carries a projection engaging in a slot in plate so that rotation of imparts a rocking movement and hence to the film-holder, which is secured to plate by means of the screw. The holder in this case is a flat one (see b, Fig. 3), and consists of a box and lid, the latter having a sector-shaped opening which admits the rays to the film. The box, and with it the film, can be rotated (by an angle of 2 /20 at a time),

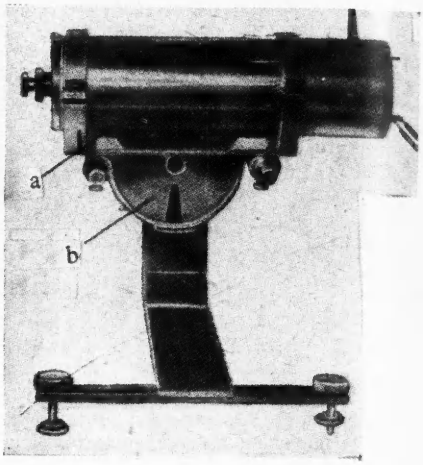


Fig. 3

Camera for large angles of reflection. (a) Oven with specimen stand ; (b) Film-holder.

while the lid remains stationary, so that successive sectors of the film are exposed to the X-rays. In this case the oven and its water-jacket have only one opening serving for both ingoing and outgoing beams. In other respects the oven is similar to that for small reflecting angles.

Recording the Breakdown Process of Austenite.

Steel specimens were heated to above A c 3, kept at that temperature until all carbides were dissolved and then rapidly moved to the hardening bath, pre-heated to requisite temperature. When the temperature of the specimens dropped to that of the bath they were moved to the oven of the camera,also pre-heated to requisite amount. There they were left until the breakdown was complete and radiographs were taken at suitable intervals to provide a record of the isothermal breakdown of austenite or of its transformation on cooling.

In order to prevent surface oxidation and the burning of carbon the heating of specimens was carried out in vacuum, under a residual pressure not in excess of 10^{-3} mm. of Hg. (otherwise the carbon tends to burn, as was established in various preliminary experiments by such means as observing the width of the resulting line).

The heating of specimens above 700° C. had to be done slowly, otherwise the liberation of absorbed gasses by the specimen and its carrier

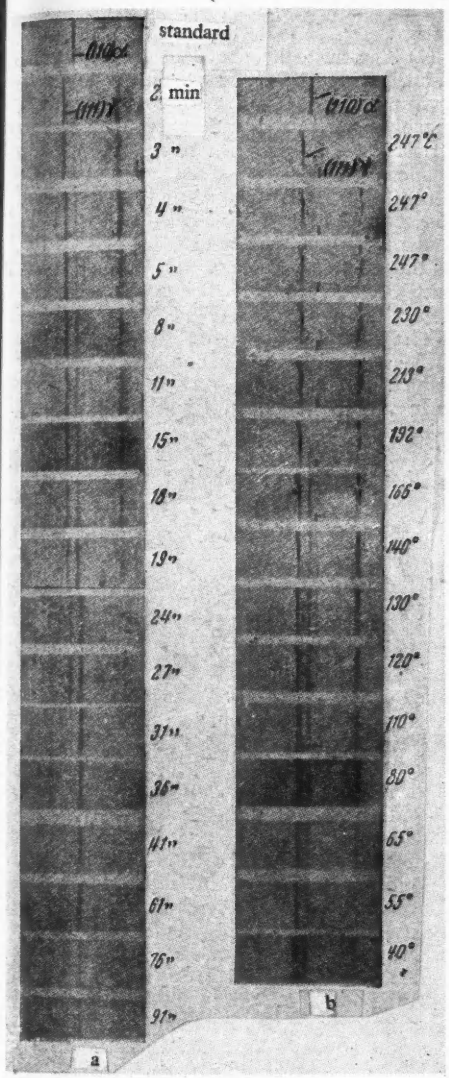


Fig. 4

Radiograph of breakdown of Austenite in 1.2% C. steel; Beam K α Cr, line (111); (a) 275° C.; (b) cooling from 250° C.

would raise the pressure to the order of 10^{-2} or even 10^{-1} mm. Hg. In order that the specimens should not lose too much heat during their transfer to the hardening bath and thence to the camera their mass has to be reasonably large. For convenience in handling, a screwed carrier

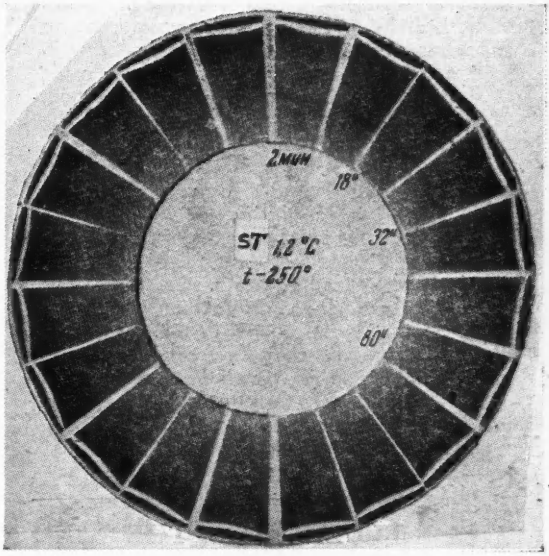


Fig. 5

Radiograph of breakdown of Austenite. Beam K B Cr, line (113).

was devised which was unscrewed and withdrawn after the specimen was in position in the camera.

To try out the new method experiments were made with 1.2% carbon steel and the breakdown of austenite was studied at temperature of 225, 250 and 275° C. The radiographs show that traces of breakdown appear after 1 hour at 225°, in 10 mins. at 250°, and in 4 mins. at 275°. A series of radiographs taken at 275° C. is given in Fig. 4a. After 2 hours at 250° and after 36 mins. at 275° the austenite lines of the radiographs disappeared. At 225° the process was not yet complete at the end of 4 hours. By comparing the intensity of lines (110)α and (111)γ and taking into account the repetition factor for these structures, it is possible, also to get quantitative data as to the development of the process. Thus, equal intensity of these lines corresponds to 60% breakdown after 2 hours at 225°, after 50 mins. at 250° and 20 mins. at 275°. The results thus obtained are in agreement with those given by magnetic measurements.

Fig. 4a shows a series of radiographs obtained from super-cooled austenite at 250° C. during further cooling to room temperature. The first few exposures show only austenite lines which are

gradually transformed into Martensite. Traces of terahedral duplication of Martensite begin to appear already at 130° C.

Fig. 5 show a series of radiographs from the camera designed for large angles of reflection. Measurements of the laltice parameter during isothermal breakdown in the region of 225-250° did not indicate any appreciable changes of condition during the early stages, while further developments could not be observed because after partial breakdown the austenite line (113) dispersed and faded into the background. This dispersion appears to be due to distortion of the laltice owing to plastic deformation consequent upon the volumetric changes in the transformation. Or it may be due to loss of uniformity of carbon content during partial breakdown.

It is now intended to conduct laltice measuring experiments with the small angle camera, where any lack of homogeneity of parameters will affect the width of lines.

The following points are particularly noteworthy:

(1) The austenite spots on the radiographs gradually change their shape during the breakdown process and become a continuous line, probably due to a re-arrangement of austenite crystals caused by their deformation due to volumetric changes.

(2) The line obtained during the breakdown process is considerably wider than that of the standard specimen (annealed at 650° C. 1.5 hrs.), and the width is dependent on the temperature at which breakdown occurs being widest with low temperatures.

The author has a certain amount of data to indicate that the width of line of the breakdown products of austenite is very similar to that of the breakdown products of martensite at similar temperature of cooling. This would seem to indicate a similarity of the structure of crystals and their process of formation.

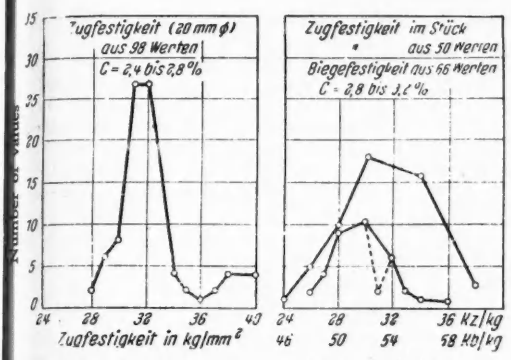
THE INFLUENCE OF THE KIND OF MELTING FURNACE ON THE QUALITY OF CAST IRON

By E. Piwowarsky, Aix-la-Chapelle. (From *Die Giesserei*, Vol. 28, No. 9, May 2nd, 1941, pp. 193-197).

Attempts were made as long as fifty years ago to produce cast iron of low carbon content and high tensile strength successfully and safely in a cupola-furnace, using a considerable proportion of scrap. A decided progress in this respect was obtained only in 1924, when K. Emmel developed his first method, the result of which can be characterised as follows: the cast iron of very low carbon content but increased silicon content allows for such large difierences of cooling rates that a preponderantly perlitic structure appears even in castings of very different thicknesses. It can be supposed that the eutectoid solidification of the fluid iron takes place when largely under-cooled and in a range of numerous nuclei. That is shown by the fine formation of graphite at the edges of the castings and by the fact that this eutectoid character remains even when the structure becomes coarser with the increasing distance from the surface of the casting.

The experience obtained with this method that the fluid product was highly overheated and had, therefore, a very fine formation of graphite led to the idea of casting a first class cast-iron of high carbon content from charges with a high scrap percentage. This resulted in the second method of Emmel, giving a product of scarcely less strength than the iron of the low carbon content, and of much higher accuracy of the intended composition as well as figures of strength. The former figures fluctuated between 2.3 and 2.9% carbon content and tensile strength between 28 and 42 kg/sq. mm. (18 to 27 tons/sq. in.), but more recently 2.7 to 3.2% carbon contents and 26 to 36 kg/sq. mm. (16.5 to 23 tons/sq. in.), could be obtained. The tapping temperature was about 1500° C., and even higher since it was no longer necessary to cast the iron without delay, transport in the foundry became of secondary importance, the shrinkage considerably smaller, and savings on risers and funnels were possible.

Based on figures given by Emmel, Fig. 1 shows the accuracy obtainable by his two methods for tensile strength and bending strength. The maxima of the curves differ only between 1 to 2 kg/sq. mm. (0.635 to 1.27 tons/sq. in.). In view of the great difference of the carbon content, this is not only surprising but at a first glance inexplicable; but Fig. 1 (a) for the older method shows a tendency of the curve to a second maximum at about 38 to 40 kg/sq. mm., as would be expected with the considerably lower carbon content of the older method. It can be assumed that this second maximum did



Old method, 1925 (Zugfestigkeit [20 mm. ϕ] aus 98 Werten ; C=2.4 bis 2.8%) = Tensile strength 20 mm. dia. taken from 98 values ; C = 2.4 to 2.8%. Zugfestigkeit in Kg/mm² = Tensile strength in Kg/mm².

New method, 1929 (Zugfestigkeit im Stück ; " aus 50 Werten ; Biegefestigkeit aus 66 Werten ; C = 2.8 bis 3.2%)=Tensile strength per piece ; Tensile strength taken from 50 values; Bending strength taken from 66 values.

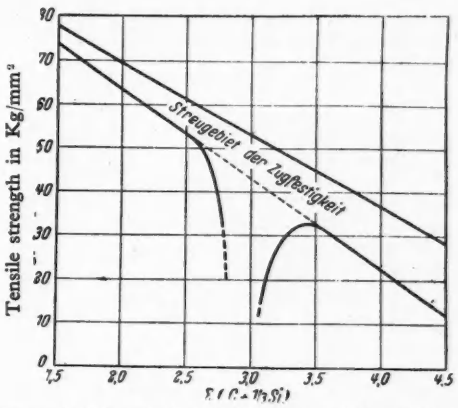
Kz/Kg = Tensile strength in kilograms ; Kb/kg = Bending strength in kilograms.

Fig. 1

not get through because of the more or less marked network formation of the graphite with its consequences detrimental to the tensile strength (see Fig. 2). This formation due to the low carbon content of the older method, could have been avoided by additions to the molten iron in the ladle of suitable high carbon irons. This has been done with success in other cases, and a tensile strength of 36 to 42 kg/sq. mm. (23 to 27 tons/sq. in.) has been obtained for cast iron with carbon contents similar to those of the older method.

It could now be verified whether, independent of the kind of the melting furnace, iron of low carbon content has really, as indicated by Fig. 2, an inclination to lower strength figures than could be expected with respect to its carbon contents. The figures of tensile strength contained in the important publications on the production of high quality cast-iron were plotted against the corresponding figures of (C + 1/3 Si) in order to afford a comparison of the various melts from the point of view of nearly the same degree of saturation.

The following is a table of the number of melts, values taken from the publications and points derived from these values :



(Streugebiet der Zugfestigkeit)=Range of the scattered values of tensile strength.

Fig. 2

Type of Furnace	Number of melts	values	points
Cupola Furnace	425	520	73
Oil Furnace	4	4	4
Air Furnace	160	160	3
Duplex Process	90	108	10
Electric Arc Furnace ..	19	19	19
High Frequency Furnace	50	138	50
Low Frequency Furnace ..	24	240	24
Total	773	1189	183

The 1189 values are mostly average figures of 2 values, about 2200 individual values being considered, but of the 160 publications on air-furnaces only 3 points have been used for the graph (Fig. 3), i.e., the highest, the lowest, and the average value. The broken lines have been entered into the graph (Fig. 3), which show, in analogy to Fig. 2, the range inside which the values of the tensile strength in dependence on the sum (C+1/3Si) should be expected. It is clear from this graph that

1. A large relapse into defective figures of tensile strength takes place between (C + 1/3Si) = 2.8 to 3.6%
2. All kinds of furnaces participate in this relapse.
3. Within the range of the expected values of the tensile strength, i.e., between the two broken lines no preference can be given to any of the various kinds of furnaces.

Two conclusions can be drawn from this knowledge, important alike to metallurgical science and to practice :

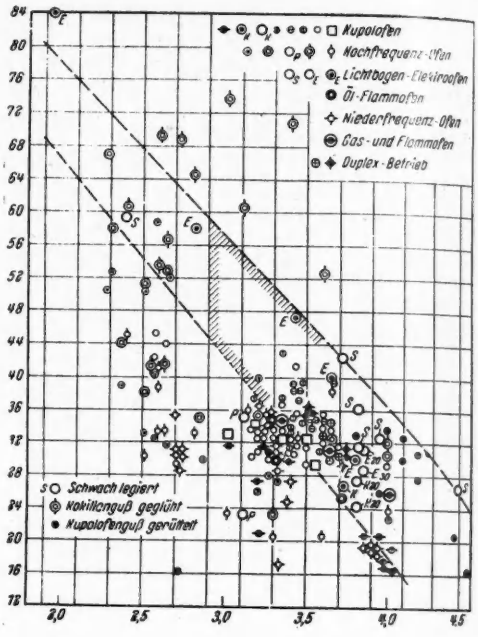
(a) The use of a certain melting furnace does not give any guarantee of the quality of the produced castings.

(b) That part of the cupola-furnace, characterised in graph (Fig. 3) by hatching, is not yet completely utilised;

in other words: apart from the production of thin castings (about below 8 mm. thickness) the cupola-furnace should satisfy the highest claims put on the tensile strength of its products if the hatched part of graph Fig. 3 would be utilised sufficiently; for it is possible to obtain a decrease of the carbon content of the molten iron in the cupola furnace with sufficient safety which guarantees a sufficient tapping and running capacity of the iron at $C + 1/3Si = 2.8$ to 3.1% (corresponding to $C = 2.4$ to 2.7% and $Si = 1.2$ to 2.1% respectively).

Certainly the melting, mixing, and refining of such classes of cast iron is by no means understood by all metallurgists; even to-day only very few high-class foundries are able to produce qualities of the kind characterised before, in the cupola furnace. The result of the investigation is, however, important in so far as it shows that expressions like electro-castings, duplex iron, etc., should not be considered as significant for cast iron qualities which could not be produced by a good cupola furnace. There is no question that cast iron of very low carbon content, i.e., up to about 2.6 to 2.8% could be produced to-day still easier in an air furnace or an electro-furnace. Such special classes of cast-iron, however, could be limited to certain less extensive purposes if the best possible class of iron was got out of the cupola furnace by using suitable raw material, careful control of the production process, and refining of the melting and casting methods. As a matter of fact some especially well managed German foundries to-day produce continuously by means of the cupola-furnace classes of cast-iron which can only be obtained in other countries, with a more favourable raw material basis, by means of alloyed cast-iron.

It can be taken for granted that the falling-off of the tensile figures as shown in Fig. 2 and 3 only happens for castings of up to 30 mm. thickness; as soon as the thickness surpasses 40 mm. the decrease of tensile strength will gradually become smaller and finally disappear completely.



(Kupolofen) = Cupola Furnace; (Hochfrequenz-Ofen) = High Frequency Furnace; (Lichtbogen-Elektro-Ofen) = Electric Arc Furnace; (Öl-Flammofen) = Oil Furnace; (Niederfrequenz-Ofen) = Low Frequency Furnace; (Gas-und Flammofen) = Air Furnace; (Schwach legiert) = low alloyed; (Kokillenguss geglüht) = chill-castings annealed; (Kupolofenguss gerüttelt) = cupola-castings joggled.

Fig. 3

Of interest is the scattering of the points in the part to the left of $C + 1/3Si = 2.8\%$ in Fig. 3. That is the case for sand-casting as well as for chill-casting. The scattering of the points is between $C + 1/3Si = 2.5$ to 2.8% more marked as in the nearly eutectoid classes of iron, i.e., the influence of the formation of graphite, or of small eutectoid quantities of graphite, on the situation of the tempering carbon after annealing, is in this case more important for the physical properties of the cast-iron, and that has to be taken into consideration, especially when casting into permanent moulds.

ELECTRIC LIGHTING IN MINES

By K. MANYGEL, Industrial Department of the Siemens-Schuckert-Werke.

(From Siemens Zeitschrift, Vol. 1940, No. May-June, pp. 120-128).

THE importance of good illumination in mines is emphasized in regard of economy and safety. The usual manner of installation is described and some types of lamps as generally used are shown. For the installation in main galleries, which are generally permanent, lead-covered cables are used, and the lamps are provided therefore with sockets suitable for such cables (Figs. 1 and 2). The lamps for the working galleries have to be installed in such a manner that alterations are easily made. Therefore rubber-insulated cables with special protective covers are used for the wiring, and the lamp sockets are more simply designed (Fig. 3). For

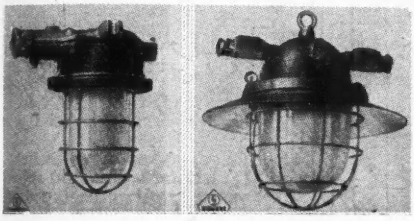


Fig. 1 Fig. 2

Lamps for permanent installations in galleries.

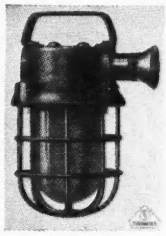


Fig. 3

Lamp for the illumination of working places.

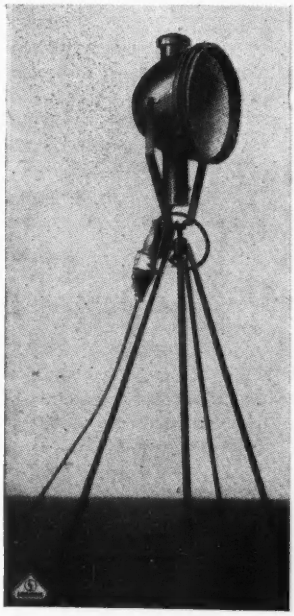
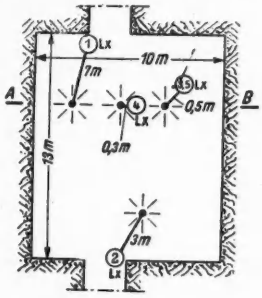
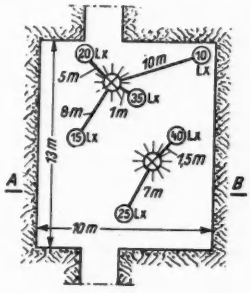
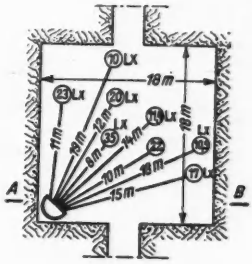


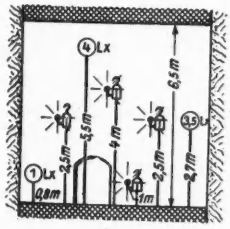
Fig. 4

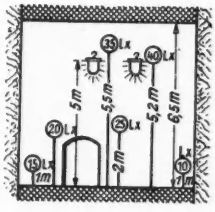
Flood-light.











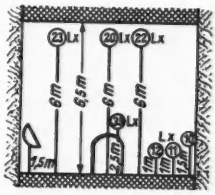


Fig. 5

Illumination with acetylene lamps.

Fig. 6

Illumination with 150 watt incandescent lamps.

Fig. 7

Illumination with flood-light using a 1000 watt incandescent lamp.

the working of seams of more than 4 m., floodlight is recommended (Fig. 4). Much better illumination can be achieved of course, by electric light than by the use of acetylene lamps. Three comparative examples are given (Figs 5, 6 and 7).

A new signalling system, combined with the lighting plant, has been developed by the Siemens - Schuckert - Werke. With signal switches inserted in the circuit of the lighting, in distances of about 20 m. (Fig. 8) signals can be given according to an agreed code by short interruptions of the light. The signals are observed in all the working galleries illuminated by the circuit. It is the special advantage of the connection used by the system, that the current of all lamps is interrupted wherever the signal switch, by which the signal is given, may be situated. The signals, which consist of extinctions of the light, can scarcely be overlooked by anybody, whatever work he is doing, whereas acoustical signals may be not heard, when the noise from drilling and other machines is loud.

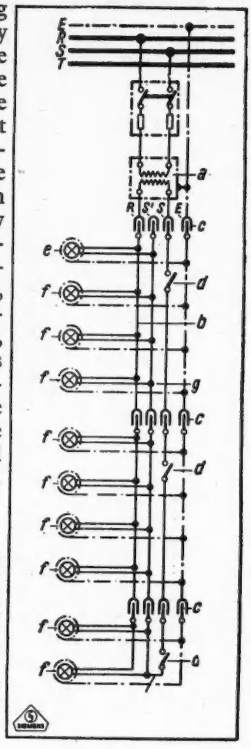


Fig. 8

Signalling system. (a) Air-cooled transformer ; (b) Rubber-covered four-wire cable ; (c) Plug socket coupling ; (d) Signal switch (in the most remoted switch the wires S and S¹ are connected) ; (e) Lamps at the place where the pails are filled ; (f) lamps along the gallery.

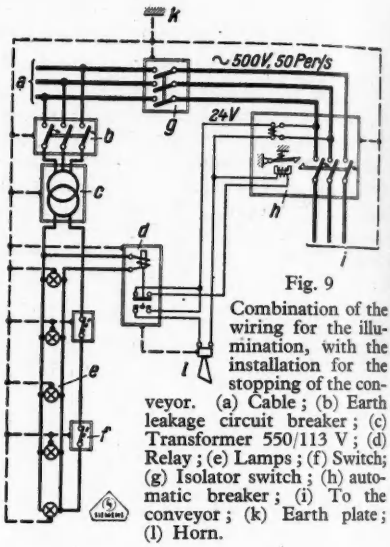


Fig. 9

Combination of the wiring for the illumination, with the installation for the stopping of the conveyor. (a) Cable ; (b) Earth leakage circuit breaker ; (c) Transformer 550/113 V ; (d) Relay ; (e) Lamps ; (f) Switch; (g) Isolator switch ; (h) automatic breaker ; (i) To the conveyor ; (k) Earth plate ; (l) Horn.

In case of interruption of the regular function of the hauling plant, all who are concerned can be informed in the shortest time, so that accumulation of coal may be avoided. It is a further advantage of the connection used that there results a lower voltage drop and therefore better lighting than with a simple two wire lighting circuit.

Fig. 9 shows another combination of the lighting plant with a signalling system. By the switches (f) the circuit of the light can be interrupted. At the same time the relay (d) is released, which opens the automatic breaker (h) and makes the horn (h) sound. In this way the motor driving a conveyor can be switched off, and at the same time all miners concerned with the working of the conveyor can be informed by the light signals. Some figures are given, proving the economic advantage of such automatic information to the shift.

BAKING PAINT WITH INFRA-RED LIGHT

By J. L. McCLOUD, Ford Motor Company, Dearborn, Mich. (Partial abstract of this publication, covering the Chapter about the use of infra-red radiation for automobile finishing. From *Industrial and Engineering Chemistry*, Vol. 33, No. 2, 1941, February, pp. 225-30).

THE Ford Motor Company, Dearborn, first used infra-red radiation for baking enamel in 1932-33. The first large-scale application was a light cave (Fig. 1) completely studded with lamps, large enough to take an automobile body (distance of lamp bulbs from body of car 12 to

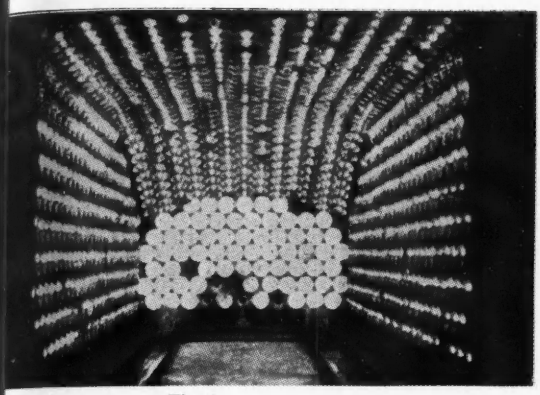


Fig. 1
Light cave for baking enamel on a completed automobile.

18 inches). It was here demonstrated, that prime paint can be baked in an extremely short time. The paints were of the oleoresinous type—a combination of natural gum resin with linseed and tung oil, pigmented to the customary low luster level of primers at that time (baking cycle of one hour at about 250° Fahr.). By addition of small amounts of lead, manganese and cobalt the company got the drying time of the primer down to 7 minutes.

Subsequently a series of clamshell ovens was installed at the Rouge plant for the baking of the prime (Fig. 2). The two halves of the clamshell surrounded the automobile body hanging on the centre conveyer line and the three together moved as a unit for the required time. This proved to be a very good method of baking the prime, but if any changes were made in the type of body hung on the conveyer, it was necessary to rebuild the conveyer. Consequently tunnels were used successfully for the baking of prime on automobile bodies (Fig. 3).

The all-over baking of the finish coat on automobile bodies is much more difficult because colour must be maintained and many colours are susceptible to the temperature differences, that are of no significance in the undercoat. Finished enamel on the bodies themselves is not baked on with infra-red except for repairs, which can be handled as individual cases and given appropriate attention.

Many other objects in automobile construc-

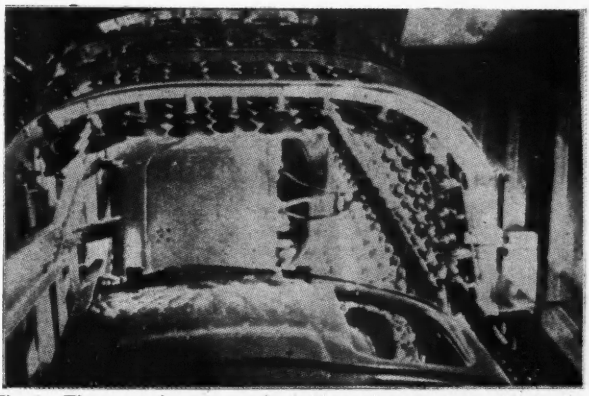


Fig. 2 First tunnel-type oven for baking prime on automobile bodies.

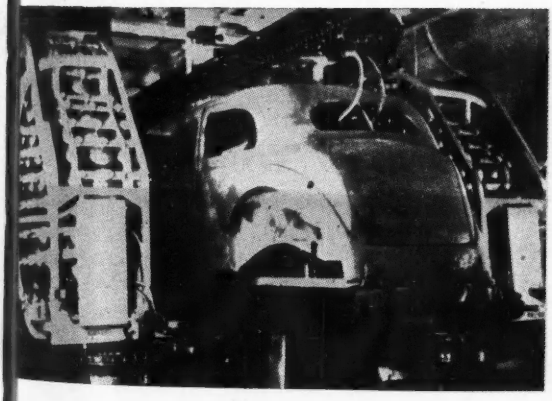


Fig. 2
View of a car-body entering a clamshell oven.

tion, especially those which are symmetrical, lend themselves ideally to baking with infra-red too. A special oven has been built for the baking of steering post columns. Prior to the installation of this oven, a one-hour bake had been given these columns in a large oil fired oven. Now the time has been brought down to 3 minutes. Wheels are baked with infra-red too. The oven has the form of a long tube of light, the reflectors of the lamps directing the energy of the lamps in toward the centre. The wheels pass the tunnel on a conveyer.

It is equally possible to use infra-red heating, when the objects are not of steel. Ford steering wheels, made of rubber over an inner core of steel, receive a high-luster smooth finish, when they are given a coating of special enamel. The time of finishing has been reduced from 2 hours (with conventional ovens) to 12 minutes with infra-red radiation. Even this time has been cut by using a modification of the enamel. The

quality of the finish has been improved in regard of the effect of perspiration from the hands on the finish.

Further applications are: Drying of latex sprayed on cloth as a cement, drying of photographic films, heating of aluminium pistons in order to expand them for the insertion of piston pins. An exceptional application was the drying of motors, armatures, etc., of the Louisville branch of the Ford Motor Company subsequent to the Ohio River flood in 1937.

It is evident from what has been said before, that an outstanding advantage of the use of infra-red radiation is the shortening of time. Another advantage is the fact that less space is used than with other methods of drying. Infra-red heating can be installed in many existing buildings without having to add to them.

The cost of baking is dependent primarily on the cost of electric power. With a power cost more than 1.5 cents per kilowatt hour the heating is expensive. The cost of the Ford Motor Company is lower, of course, and the method is economical for this factory.

TESTING OF CRANK SHAFTS

By E. SIEBEL and G. STAHLI. (From *Die Giesserei*, Vol. 28, No. 7, April 4th, 1941, pp. 145-150).

THE crank shaft of a combustion engine is subject to an alternating stress, produced by the gas pressure in the cylinder and the inertia forces of the reciprocating parts. At high speed the stress in the crank shaft is predominantly influenced by the inertia forces. Therefore, it appeared convenient to design a testing machine in which various crank shafts could be examined under the action of an alternating inertia force. The problem can be easily solved by supporting a crank shaft in a similar manner to that in an actual engine and attaching weights to a carrier taking the place of the piston. The crank shaft is rotated by a coupled motor. In this way it is ensured that conditions are very much the same as in an engine, and the method has the further advantage over testing in an actual engine that the speed and loading can be varied within wide limits.

The schematic arrangement of the testing machine is shown in Fig. 1. The crank shaft (a) is driven by a motor (h) through a coupling (f) and a fly-wheel (g). The carriers (i) are subject to a reciprocating motion between the guides (c) and carry small weights (d). (b) is the connecting rod. Since the speed of the crank shaft in the testing machine is only about half that of an engine, the connecting rods can be so strongly designed that no fracture will occur in them. To damp out vibrations, the machine is mounted on a concrete block which is placed on a rubber carpet. Ample lubrication to all moving parts is provided, since the bearings are exposed to great stresses. The driving power is provided by an A.C. motor, which has a maximum speed of about 1470 r.p.m., the speed being indicated on a tachometer.

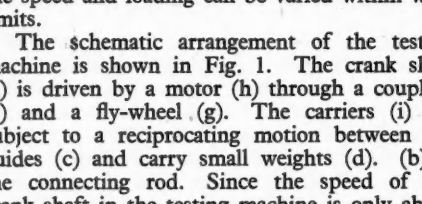


Fig. 1

Investigations were carried out on crank shafts used by the Wanderer W. 24 four cylinder engine.

Comparative Conditions of Loading in an Engine and in the Testing Machine.

The forces, due to gas pressure and inertia, acting on a crank shaft are shown in Fig. 2. In addition the crank pin and part of the mass of the connecting rod (considered as concentrated at the crank pin) constitute a constant centrifugal force. The resultant of all these forces acting on the crank shaft is shown in Fig. 3. It can be seen that the magnitude of the force varies between + 1420 and − 150 kg, and the maximum is reached when the piston is at the dead centre position.

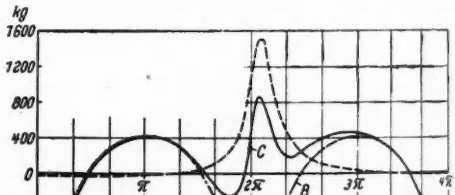


Fig. 2

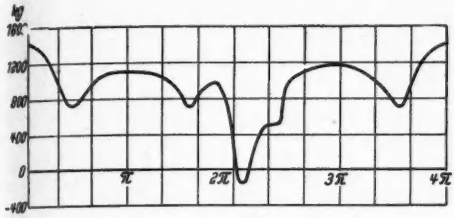


Fig. 3

The turning moment of a single cylinder exerted on the crank shaft is shown in Fig. 4. The resultant turning moment of all four cylinders at the coupling is shown in Fig. 4, by the line d. It can be seen that the turning moment changes sign twice during each revolution and lies between the limits of + 50 and −27 kg/m. Unlike the force, the maximum bending moment occurs when the crank is at 45° to the dead centre position.

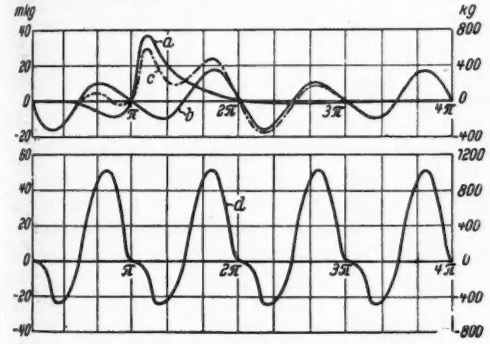


Fig. 4

A special case arises when the engine is idling. In this case the turning moment of the inertia forces is about 50% greater than that at full load, since in the latter case the gas pressure opposes the turning moment of the inertia forces.

Returning to the testing machine, it is obvious that conditions here are similar to that of an idling engine, since gas pressure is completely absent. The cyclic variation of the inertia forces acting on the crank shaft at 1470 r.p.m. is shown in Fig. 5. The following weights formed the basis of the calculations: weight of carrier and pin 7.45 kg, weight of connecting rod 4.75 kg. The reciprocating part of the connecting rod was assumed to be only 1/6th of the weight of the connecting rod, owing to the heavy big

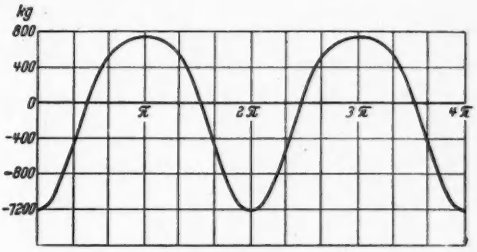


Fig. 5

end. These forces when superimposed on the centrifugal force give a resultant shown in Fig. 7. Its magnitude varies between + 1730 and + 515 kg. Thus the variation is 1730−515 = 1215 kg, which is almost the same as that of an engine at full load.

It should be noted that the loading of the bearings in the testing machine is about 50% higher than that of an actual engine, and that the stress is never reduced to zero (compare

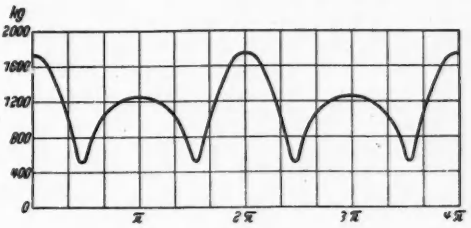


Fig. 6

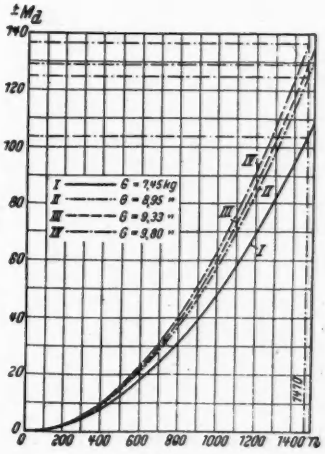


Fig. 7

with Fig. 3), owing to the complete absence of gas pressure.

Finally, in Fig. 7 the turning moment of one cylinder and that of all four cylinders are shown. The turning moment varies between ± 99 mkg. This means that the loading of the crank shaft end is about 1.8 times that of an actual engine at idling. The turning moment of the electric motor to overcome friction in the bearings is not taken into account.

The stresses in the crank shaft at other speeds than 1470 ($= n_0$) r.p.m. are in the ratio $(n/n_0)^2$, and the stresses at other

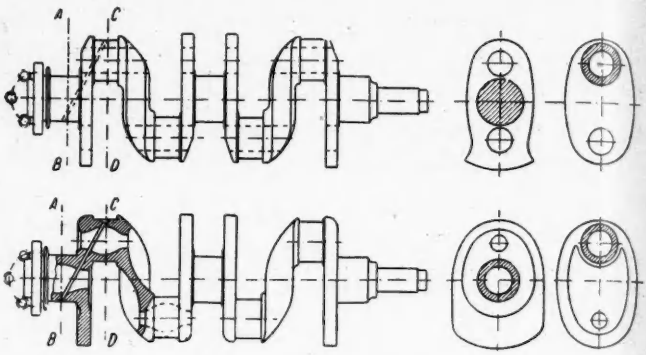


Fig. 9

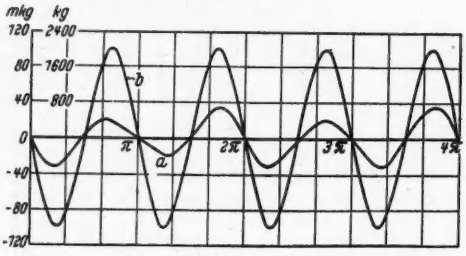


Fig. 8

loading of the carriers are simply in the ratio of the weights G/G_0. In Fig. 8 the stresses are plotted for various speeds and weights on the carriers.

Experimental Results.

Investigations were carried out on cast and forged steel crank shafts, Type Wanderer W.24 (Fig. 9), their resistance to alternating inertia forces determined and compared. The low carbon content cast crank shafts had an average tensile strength of about 27.5 kg/mm², whilst those of the forged steel type had 86.8 kg/mm². The shafts were carefully embedded in the apparatus and the speed was increased by 100 revs. every 10 minutes. The speed was then kept constant until fracture occurred. The results are given in Table I. The nominal stress represents the turning moment at 1470 r.p.m.

(To be continued)

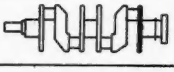
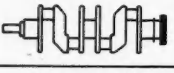
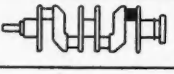
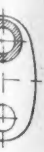
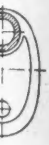
Cast Iron Crankshaft Type W 24				*Steel Crankshaft Type W 24*			
Nr.	*± Md. mkg*	*Revs. until Fracture*	*Place of Fracture*	*Nr.*	*± Md. mkg*	*Revs. until Fracture*	*Place of Fracture*
13	104	4 238 000	No Fracture	1	104	2 016 300	No Fracture
6	104	4 506 000		1	125	2 795 000	
14	104	540 000		2	130	5 000 000	No Fracture
18A	125	989 000		3	136,5	4 807 000	No Fracture
8	130	Few					
18B	130	Few					

Table I





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